EXHAUST THIMBLE FOR ARCTIC ENVIRONMENTS

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EXHAUST THIMBLE FOR ARCTIC ENVIRONMENTS

A

PROJECT

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By

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Abstract: The purpose of this research project is to investigate an alternative exhaust thimble design. Exhaust thimbles provide building code mandated protection of the structure materials from the heat produced by flue gases. Current designs are ill suited for arctic conditions. They do not maintain necessary building envelope and thermal insulation integrity. A well thought out arctic design would allow for the thermal and vapor barrier integrity to be maintained without sacrificing performance. Design concepts were rendered by engineers at UAF due to necessity. From concept, full scale models were built and tested in summer and winter conditions. These prototypes use a natural convective process to maintain an outer layer skin temperature below the combustion range. Thermocouples were placed to capture the transient and steady state thermal data and a hot wire anemometer was used to record flow velocities at steady state. Following field research, the collected data was organized and used to refine computer modeling that was done using COMSOL software. The result was a clear indication that this design has promise.

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Introduction: Thimbles provide a means for flue gas piping to transition through building envelopes, whether they are roof or wall, safely to exhaust to the exterior environment. The thimble insulates the surrounding building materials from the exhaust gas heat and protects the structure from fire and heat damage, For this reason, they are mandated by Fire and Safety Codes nationwide. The general practice for residential and light commercial burners is to use insulated exhaust stack piping and run it through some sort of additional air spacing box at the building envelope like in Figure 3, since the insulated stack is not UL zero clearance rated. These boxes are vented for draft cooling. Commercial systems might use something similar to what is seen in Figures 1&2. With the systems in Figures 1&2, the air near the exhaust pipe is heated and rises due to buoyant forces. This creates a pressure differential between the inside and outside and results in cooler outside air flowing in through the outer annular ring in order to balance the pressure.

Background: Dealing with exhaust gases, whether they be from a wood stove or a diesel generator can present real issues in northern latitudes. Building designers are faced with the task of properly implementing fire prevention methods and still maintaining the building envelope integrity. This can result in cold spots and vapor barrier breaks that can in turn develop frost issues and reduce the overall building efficiency. If proper spacing of combustible material from the exhaust thimble is not allowed, fires can and do persist. Typically what happens is the burner comes on and runs as intended and for some reason either begins to run at higher temperatures or runs for a longer duration than intended. The heat from the flue gases overcomes the insulation and begins to heat the surrounding combustible materials. This can happen once and result in flame or happen over years and slowly cinder the building materials; either way it can and does cause tremendous and costly damage to the building and property inside. In the arctic many village/small grid communities rely on diesel gensets to provide electrical power and in this case many powerhouse fires have occurred due to runaway gensets and prolonged exposure of combustible materials to cindering temperatures.

Research suggests that a better thimble design could help to alleviate these issues by dealing with the heat transfer more proactively while also maintaining the building envelope. While designing and constructing the Energy Technology Facility on the University of Alaska Fairbanks campus, engineers developed a plan for a more suitable arctic thimble. These design engineers along with research engineers at UAF worked together to develop a proposal for the Alaska Emerging Energy Technology Fund. It was proposed that using a natural convection fresh air thimble will reduce initial costs, allow for tighter building designs around exhaust, and deal with runaway and prolonged flue gas temperatures better than any thimble on the market today. The design is simplistic in appearance, Figure 4, and work was to be conducted to find optimal geometric sizes for future prototypes. From Figure 4 it is seen that the thimble uses two annular spaces for cool fresh air to enter, remove heat from the stack and, as will be seen in this report, keep the surface temperatures at the envelope well below the UL listed combustion specifications, approximately 160°F. This system has great potential especially in the arctic, in that as the demand for heat and power increases during the winter months the ΔT of the cooling air also increases, thereby increasing the efficiency of the overall system. This dynamic heat handling design will be simplistic enough in its design and construction that it should have no problem replacing the vast majority of traditional exhaust thimbles, if not all.

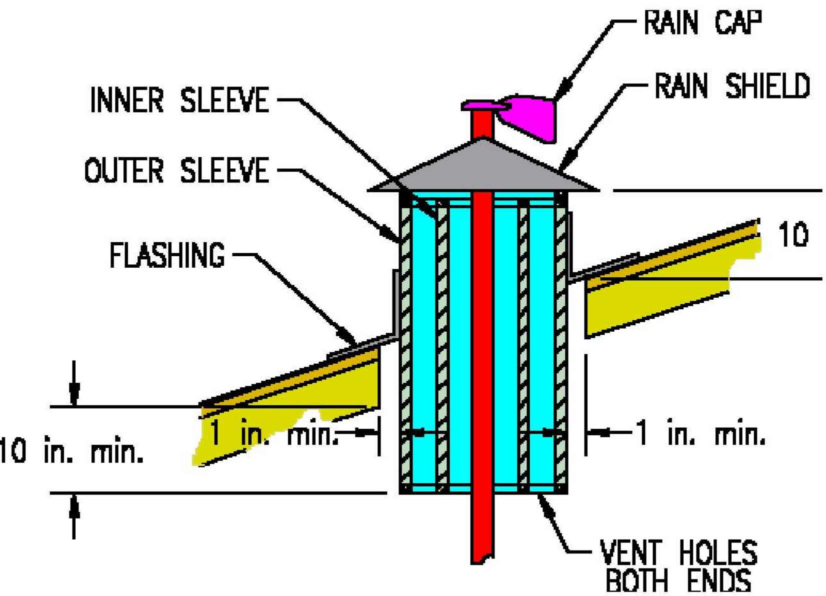

Figure 1: Traditional Commercial Roof Exhaust Thimble

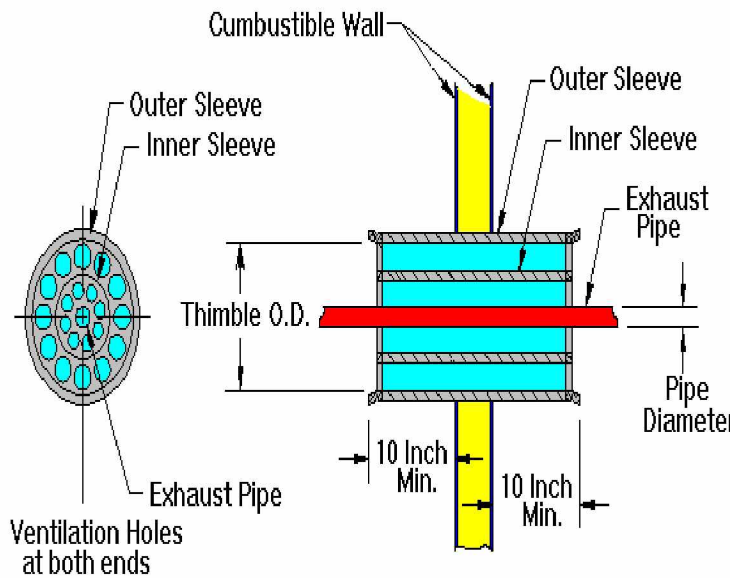

Figure 2: Traditional Wall Exhaust Thimble

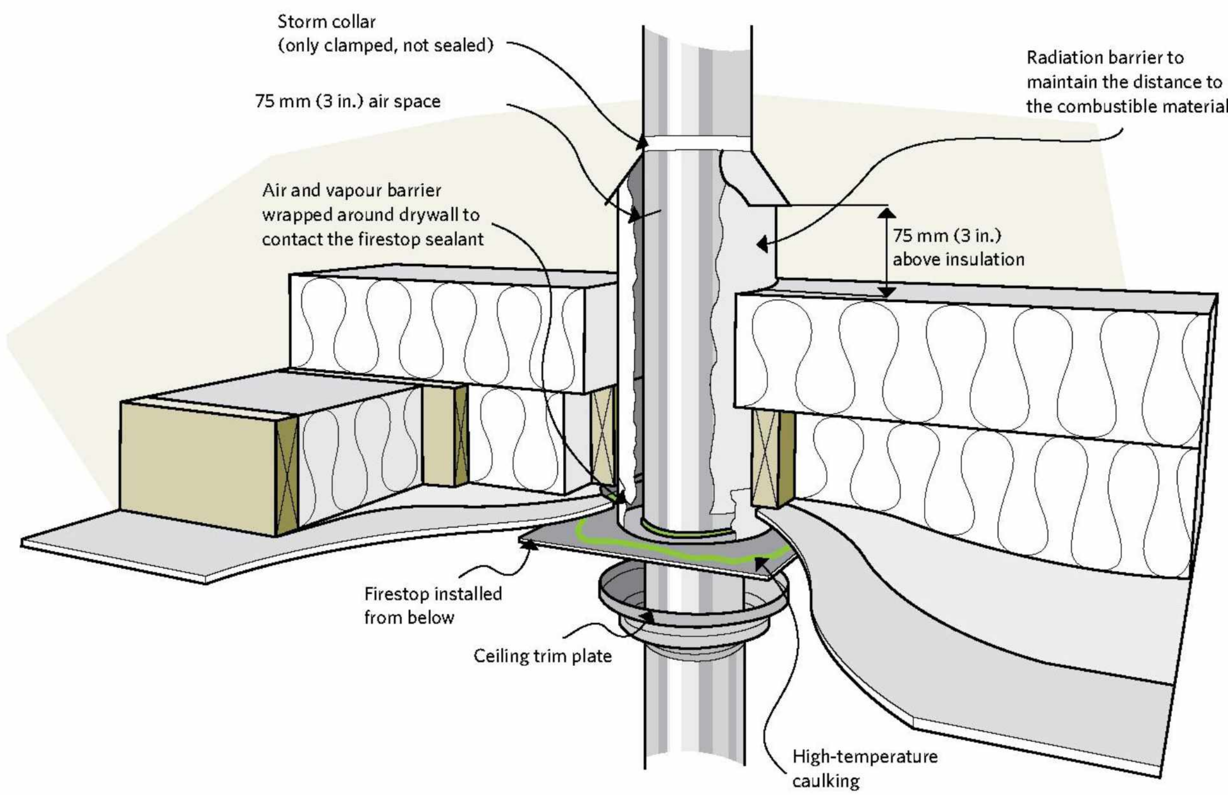

Figure 3: Traditional Ceiling Exhaust Thimble

Equipment:

A	B	C
2”	3”	6”
4”	5”	8”
6”	7”	10”
10”	11”	14”

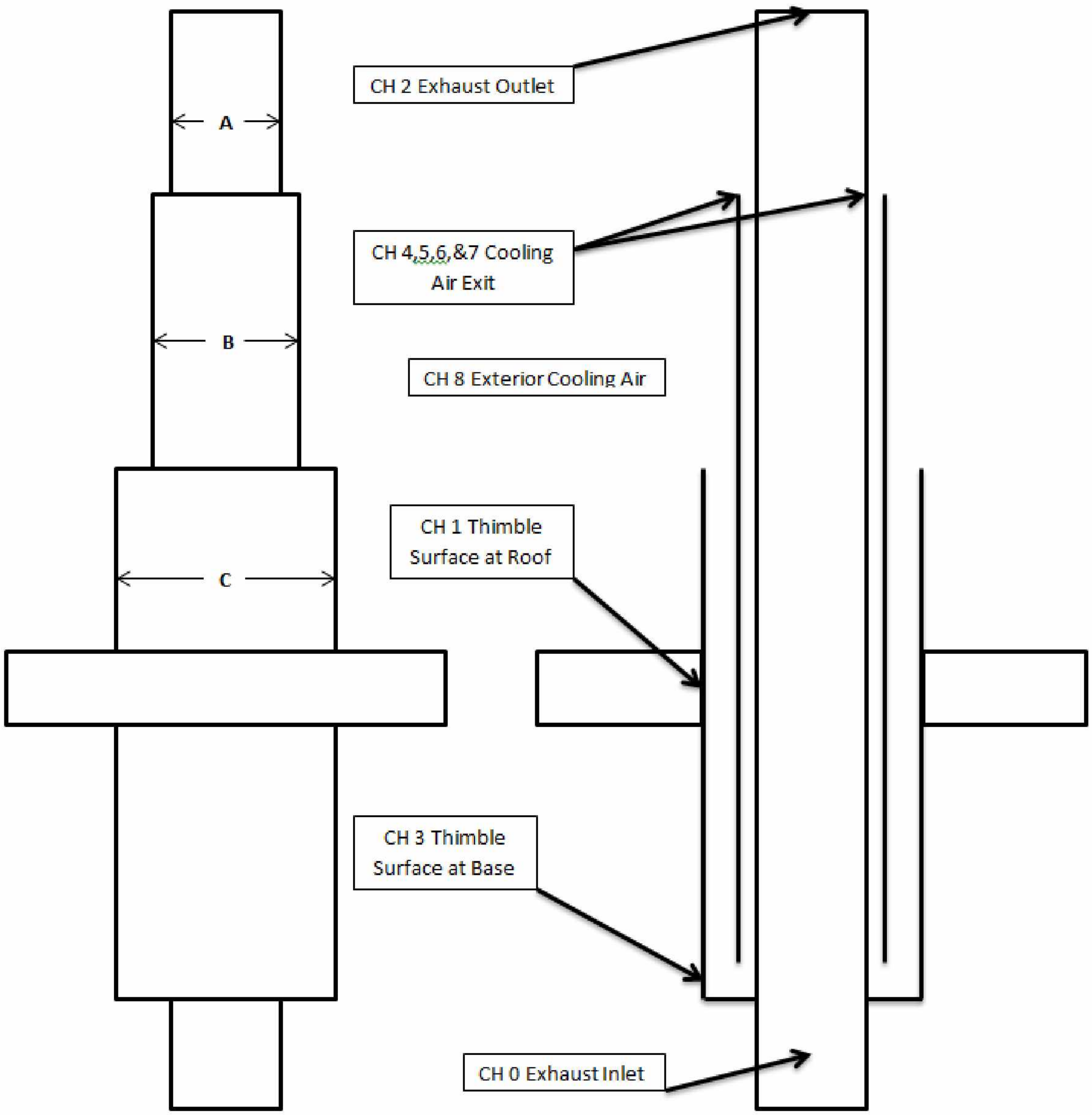

Figure 4: Sketch of Exhaust Thimble Design

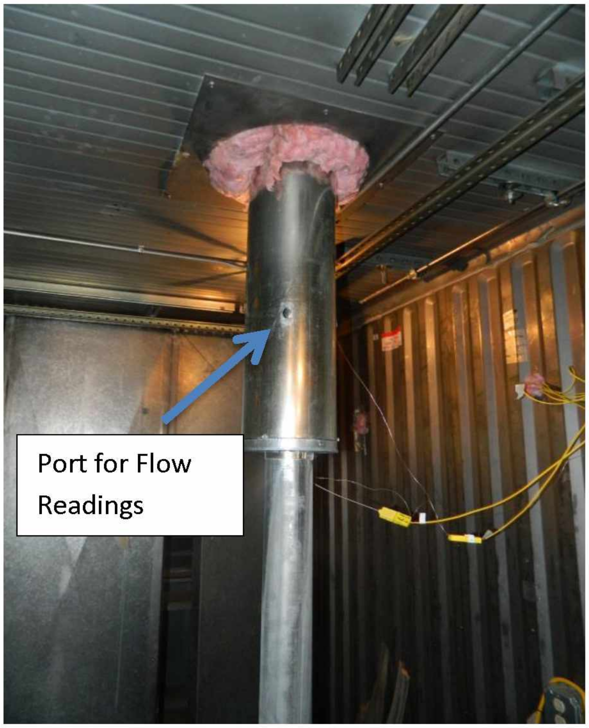

Figure 5: 4" Thimble Installed for Testing

Figure 7: 10" Thimble

Figure 6: 2" Thimble Stored for Future Tests

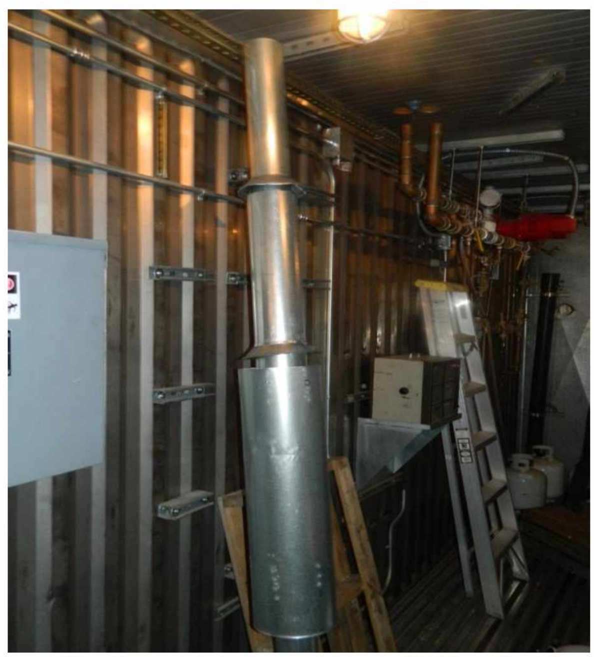

Figure 8: 6" Thimble

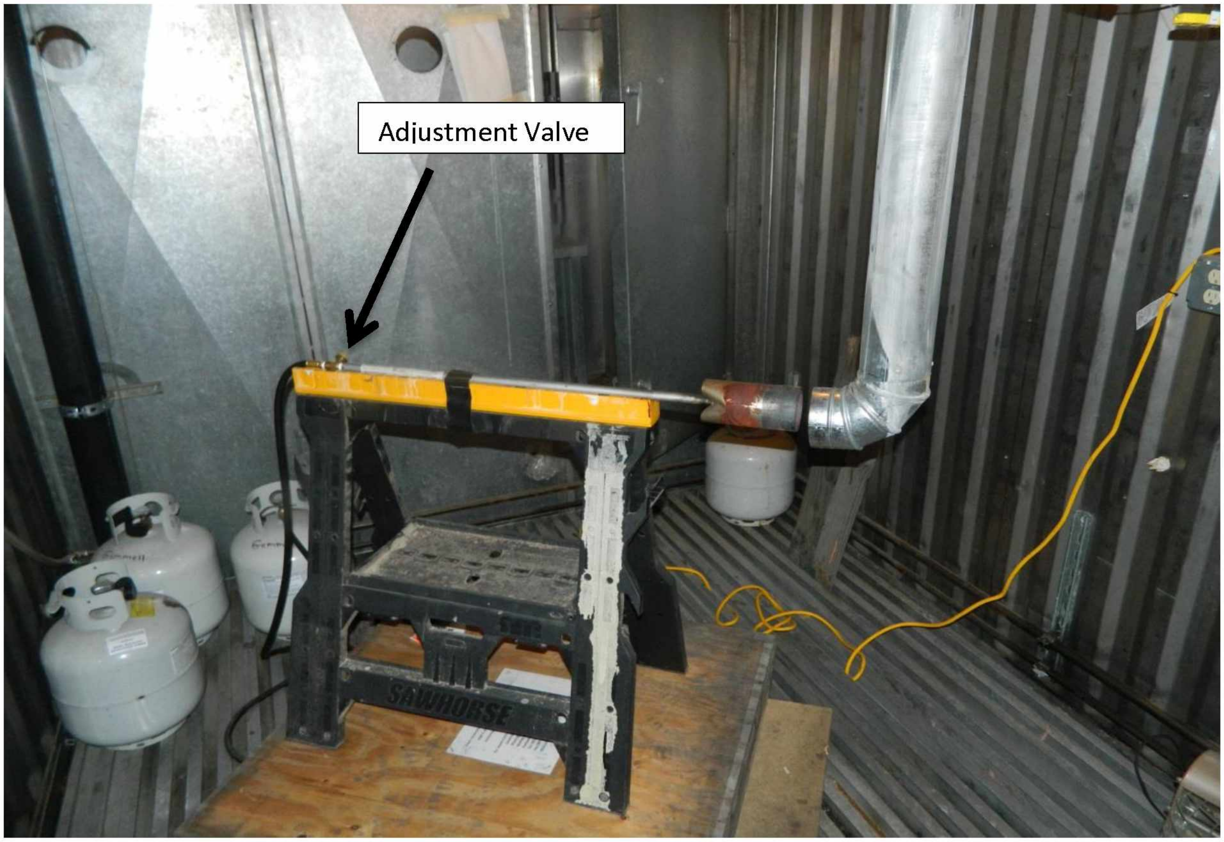

Figure 9: "Weed Burner"/Adjustable Heat Source w/Propane Tank

Figure 10: Hot Wire Anemometer

Procedure/Performance: For testing, single wall nominal sized exhaust stack piping was used for construction of the thimbles as can be seen in Figures 5-8. All tests were conducted in a repurposed refrigeration connex on the UAF Campus directly behind the MIRL Building. For the heat source, a "weed burner" Figure 9, was used to combust propane and vent it up the stack of the various thimbles. The flow rate was adjusted to reach target stack temperatures of 400°F, 600°F, 800°F and 1000°F. Multiple tests at each temperature and condition were run to ensure an appropriate sample. Thermocouples were prepositioned on the thimble to record various temperatures over the duration of each test, see Figure 4; i.e.: exhaust inlet/outlet, cooling air inlet/outlet, envelope surface, and thimble base skin temperatures. National Instruments Labview Software and Data Loggers were used to collect and tabulate all of the data. Data was later transferred to Excel for manipulation and plotting. As the skin of the flue gas piping heats up, buoyant forces will force the air in contact with this surface to rise and induce a draft drawing ambient cooling air into the outer annulus through the inner annulus and back into the environment drawing heat away from the system. With any heat exchanger it is important to know fluid flow, so this value was captured under steady state conditions. All tests approximately reached steady-state by the 20 minute mark, so this time marker was used to pull air flow data using the hot wire anemometer, Figure 10, when possible. Initially attempts were made to collect the velocity data using a digital monometer, but the velocities were too low and fluctuating too much to get accurate readings. This prompted the use of a digital anemometer that was capable of capturing low velocity readings. Following field testing, data was processed in Excel and a computer simulation model was built using COMSOL Software. This is Multiphysics software, of which laminar fluid flow and conductive and convective heat transfer physics were used, and with the field data the model can be modified to prove the collected steady-state values. Since the major concern is what is occurring at the steady state, transient analysis was not performed. Once a model can be built that provides results that corroborate the experimental ones, then work can be done to optimize the geometry for the best overall performance. With that information a prototype could be put through the same testing and possibly end in a marketable product within just a few iterations.

Results: As was mention previously, data was collected during both the summer and winter months. The objective was to obtain experimental results with at least a 100°F ΔT. For the summer temperatures the goal was temperatures greater than 70°F and winter temperatures were hoped to be at least -40°F or cooler. Unfortunately the summer weather during testing was rainy and cool and the following winter was mild and relatively warm, so testing was conducted at approx. 60°F in the summer and approx. -20°F in the winter. Following is the summer and winter data followed by some COMSOL models demonstrating what is happening inside the thimble. The samples are of different size thimbles at the incremental temperatures, but are representative of the performance for all thimbles at the respective temperatures.

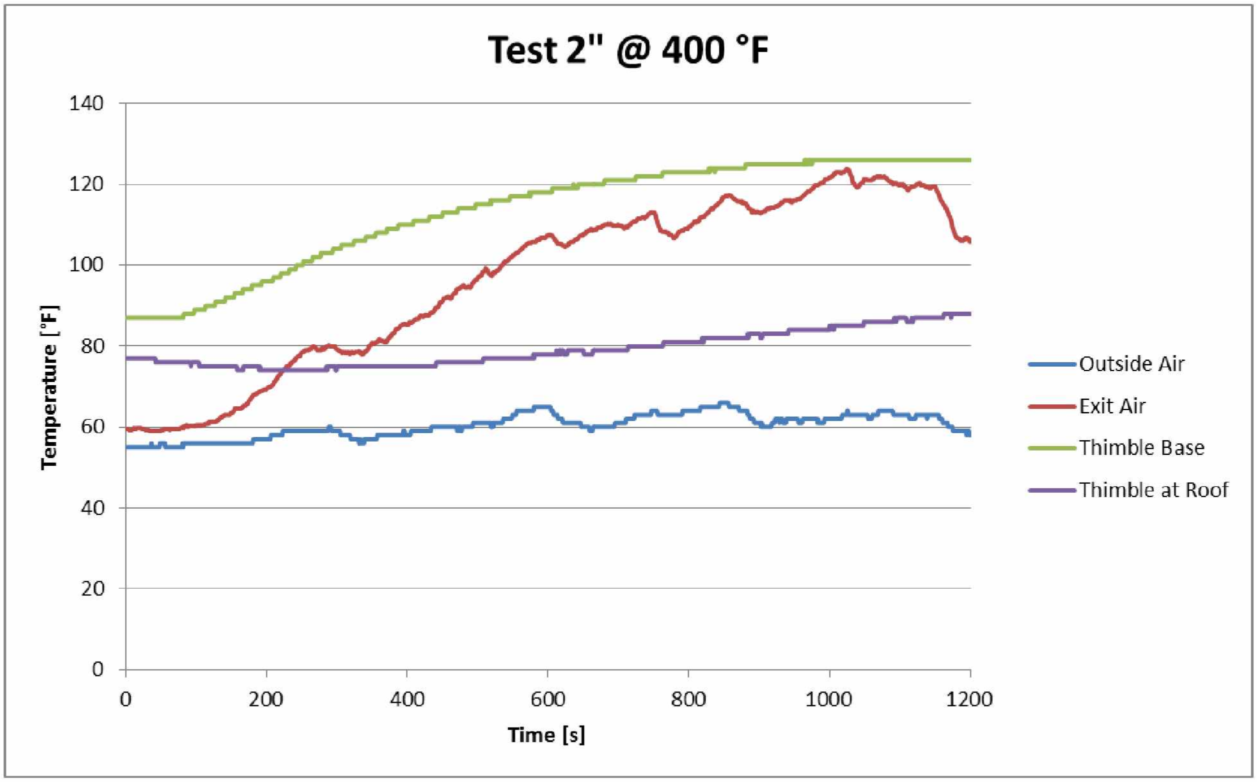

Figure 11: Summer 2" Thimble Test at 400°F

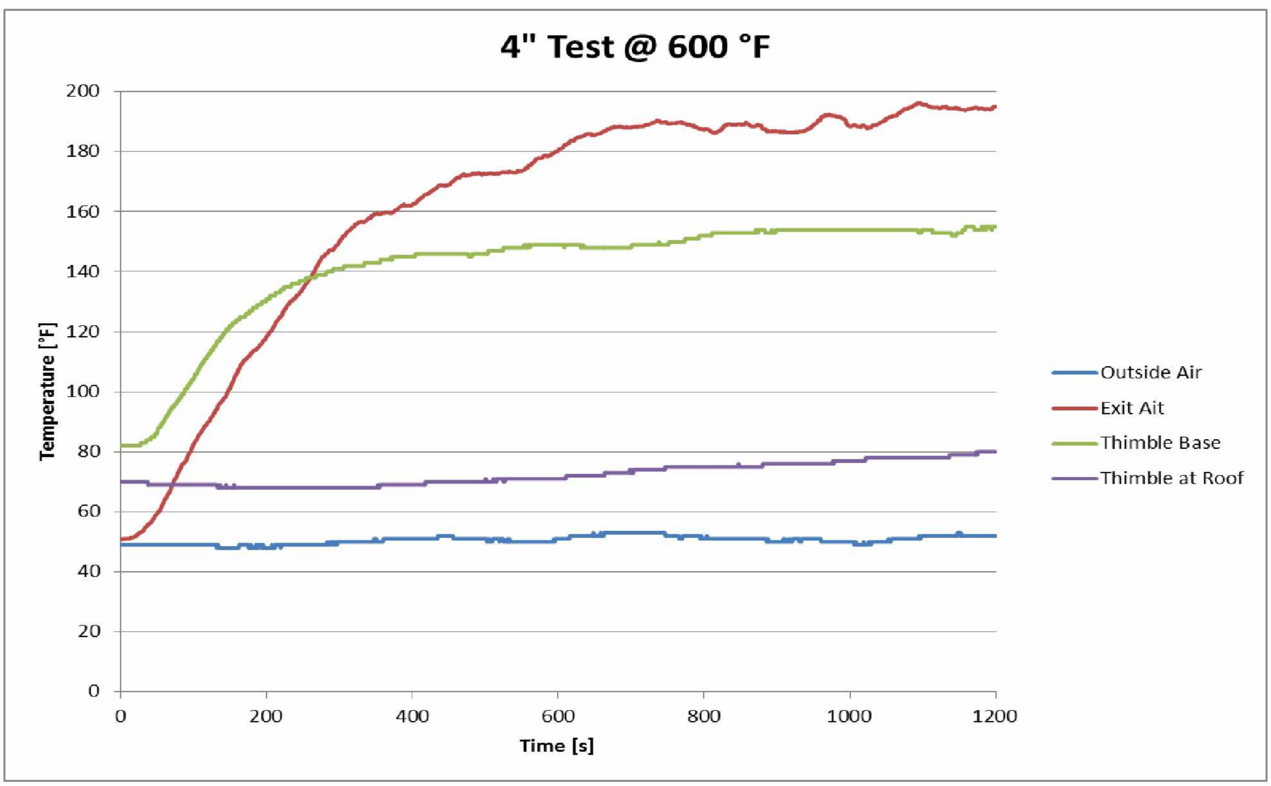

Figure 12: Summer 4" Thimble Test at 600°F

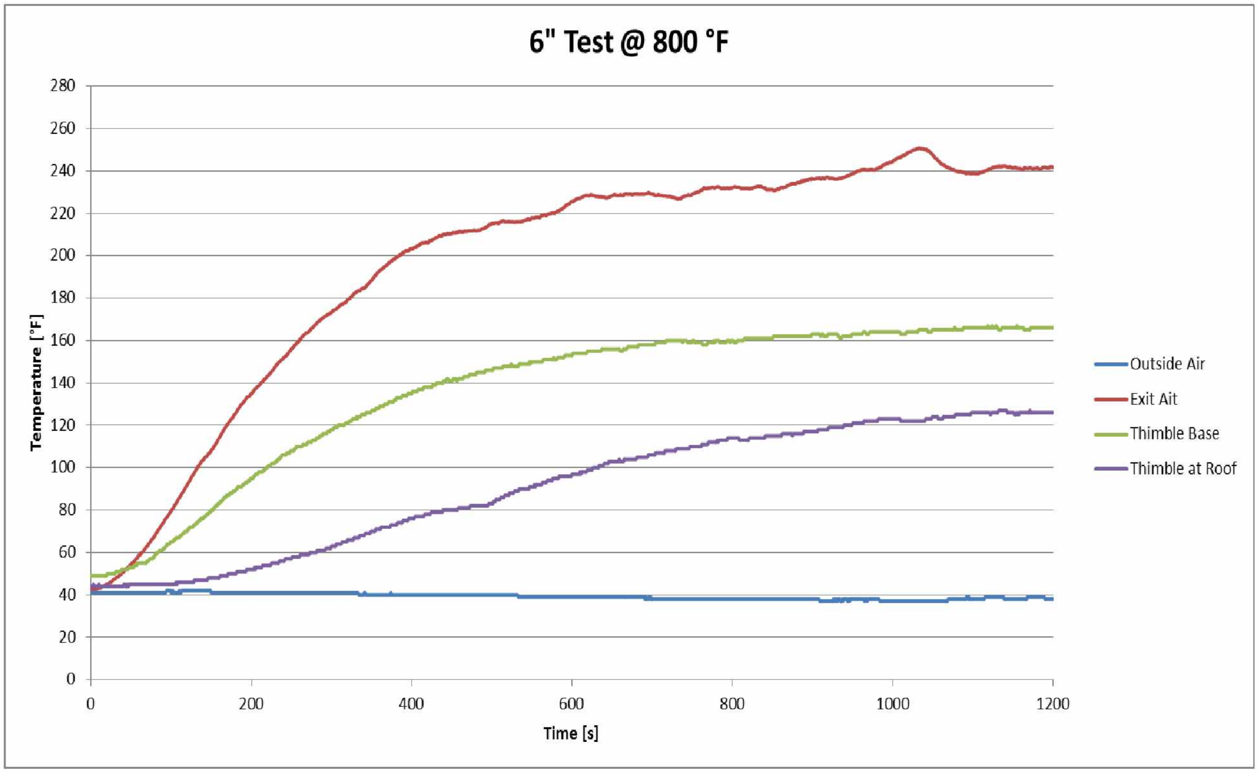

Figure 13: Summer 6" Test at 800°F

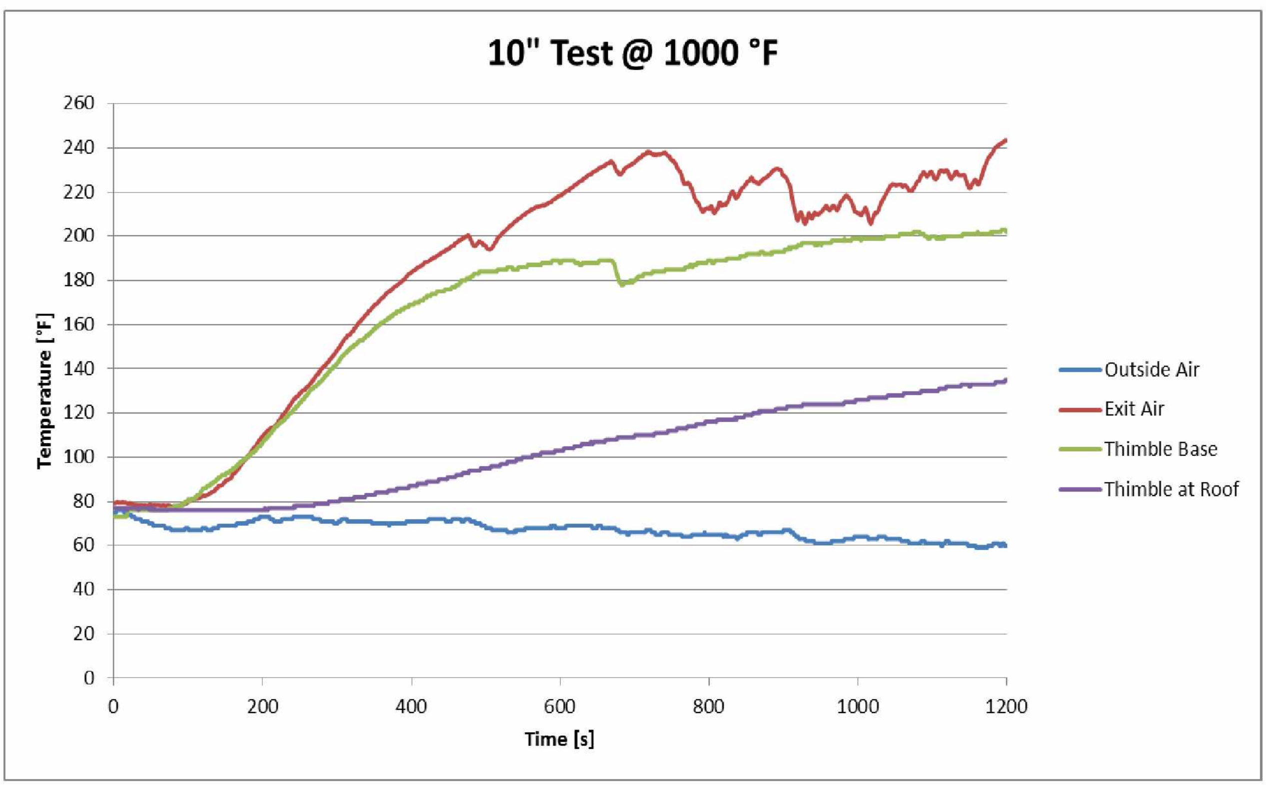

Figure 14: Summer 10" Test at 1000°F

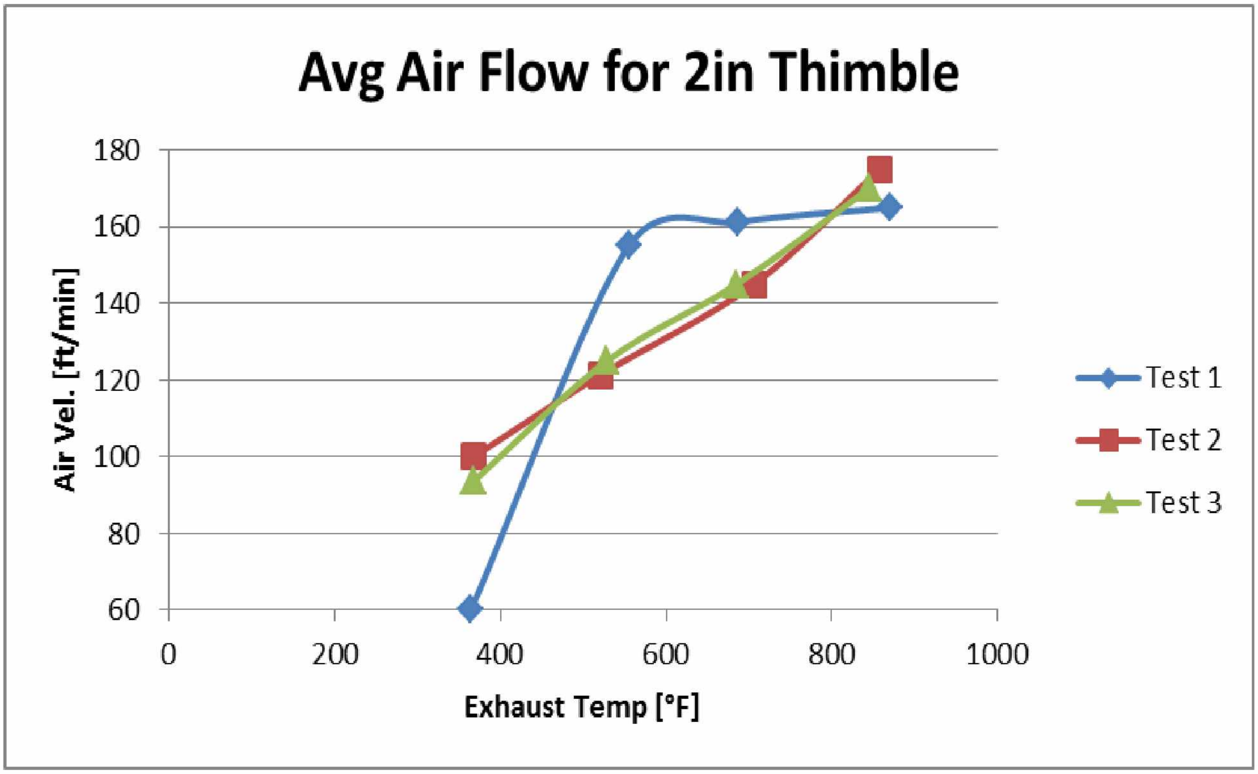

Figure 15: Cooling Air flow approximately in the center of the outer annulus

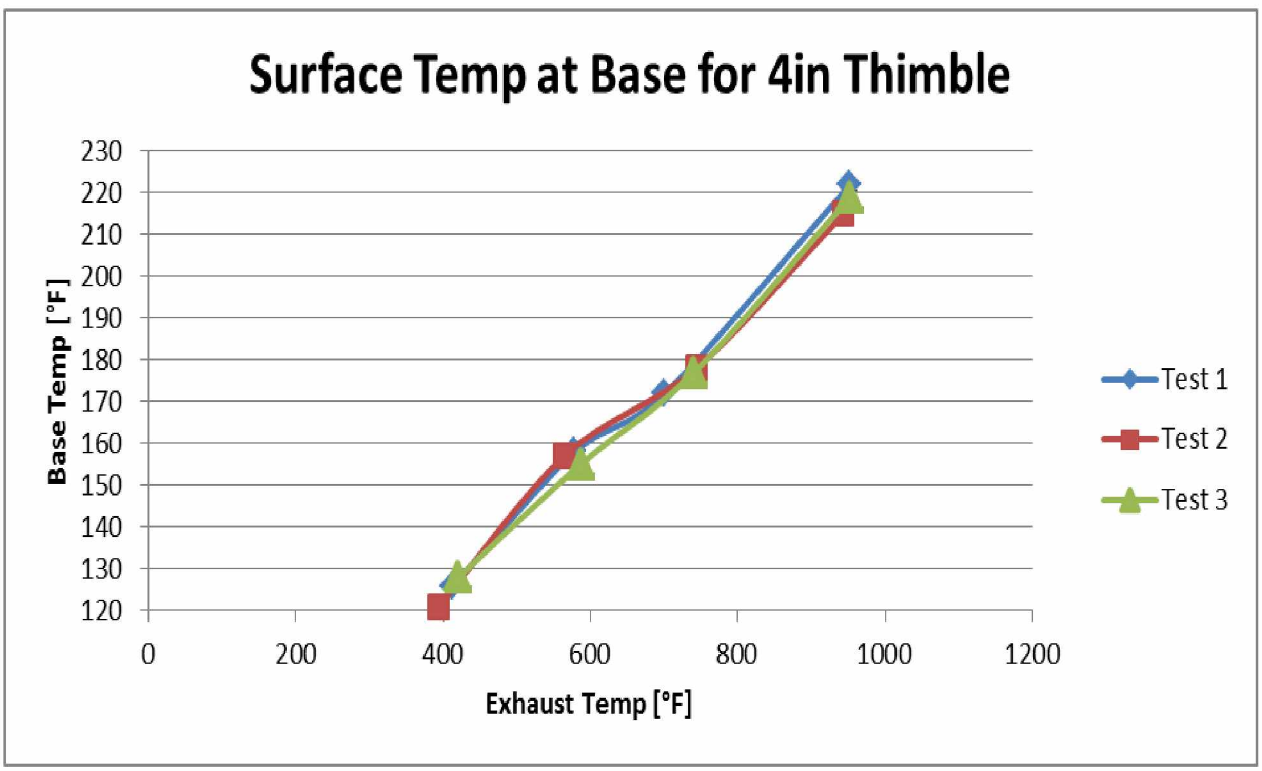

Figure 16: Temperature of the exterior skin at the base

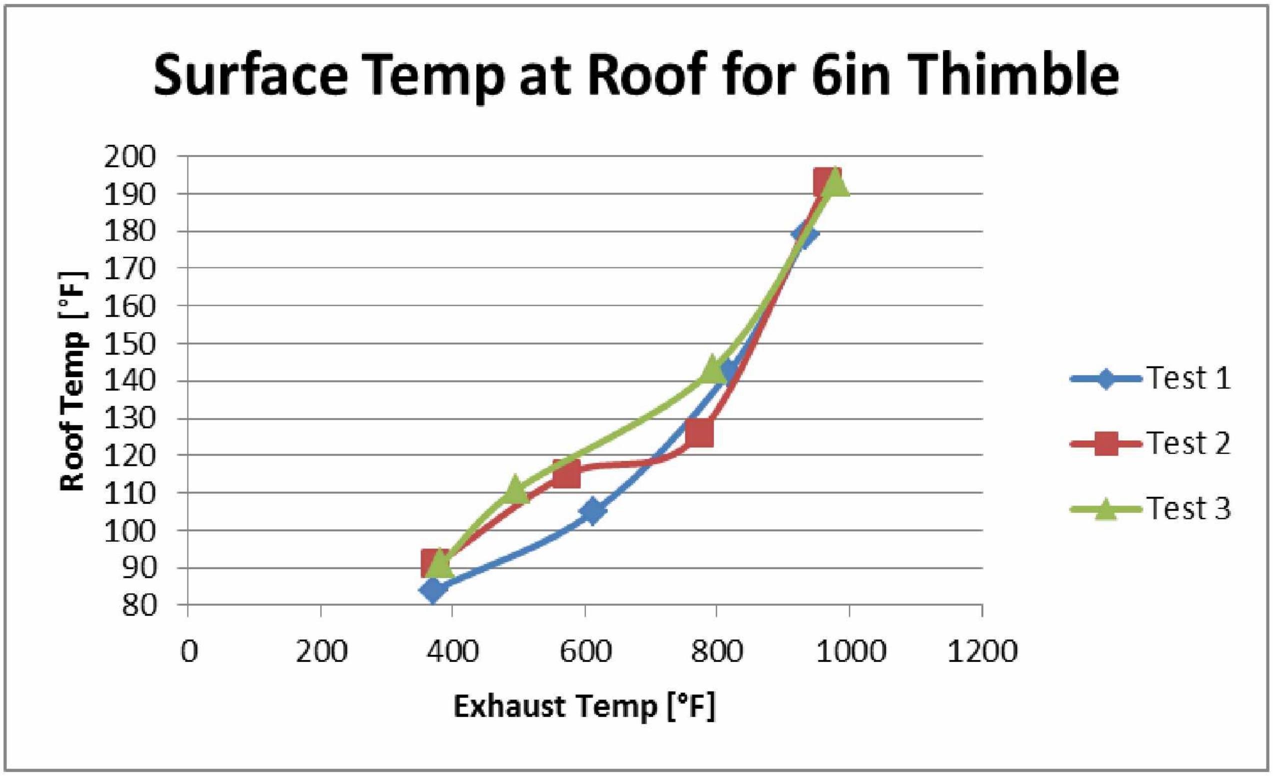

Figure 17: Temperature at the roof material and thimble interface

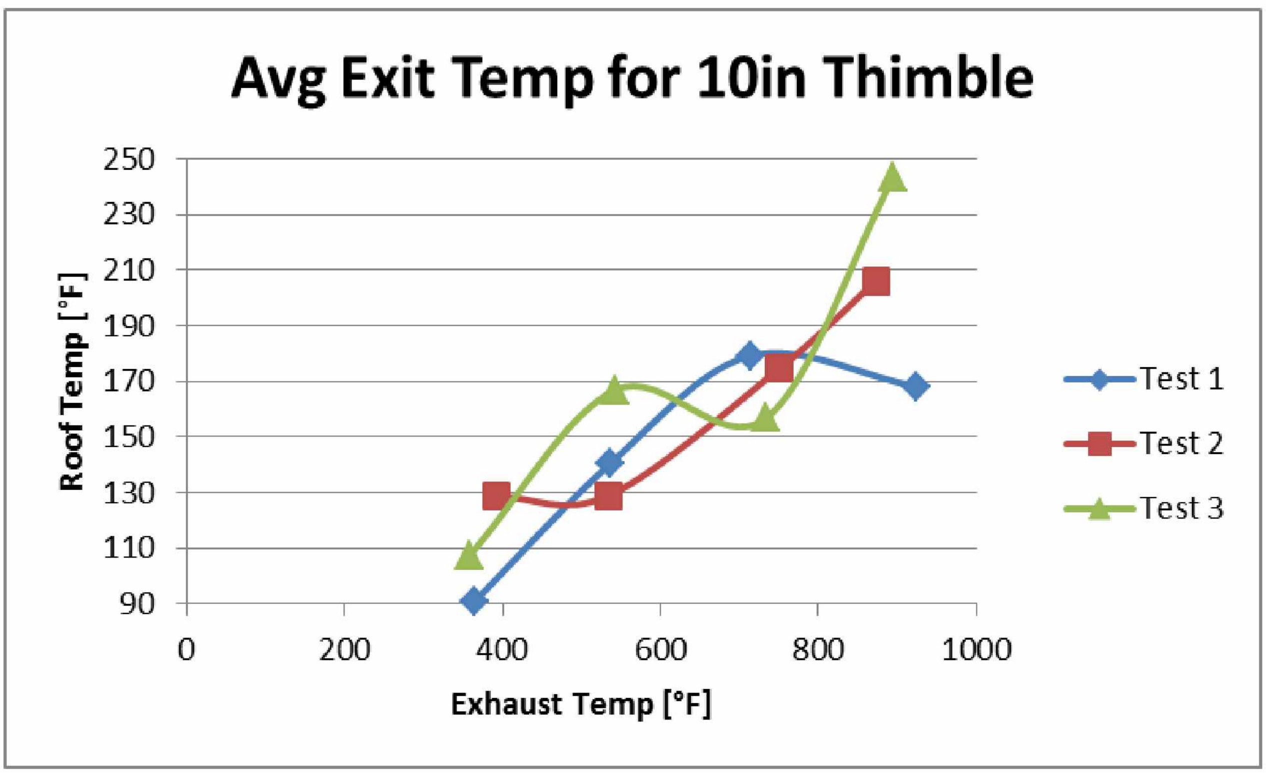

Figure 18: Average exit temperature of cooling air from inner annular ring

Figures 12-15 are good examples of the performance of the four thimbles at appropriate temperatures during the summer months. From these four graphs it can be seen that the thimble starts drawing heat away within the first 2 minutes on some designs and almost immediately on others. Figure 16 displays the recorded steady state air flow velocities of the 2 inch thimble under summer conditions. These results are representative of what was seen in the other thimbles as well under similar conditions. The upward trend of the velocity with respect to the change in temperature is appropriate. Figure 19 further affirms this with the increasing temperature of the cooling air exhaust. This demonstrates that the thimble design is able to respond to higher flue exhaust temperatures with increased cooling. Figures 17&18 show the skin temperatures of the thimble at the base and in the roof area where the thimble might be in contact with combustible building materials. From Figure 17 it is seen that the base can get quite hot, but just 18 inches above this point the temperatures are much cooler and in most cases below the charring temperatures of most building materials.

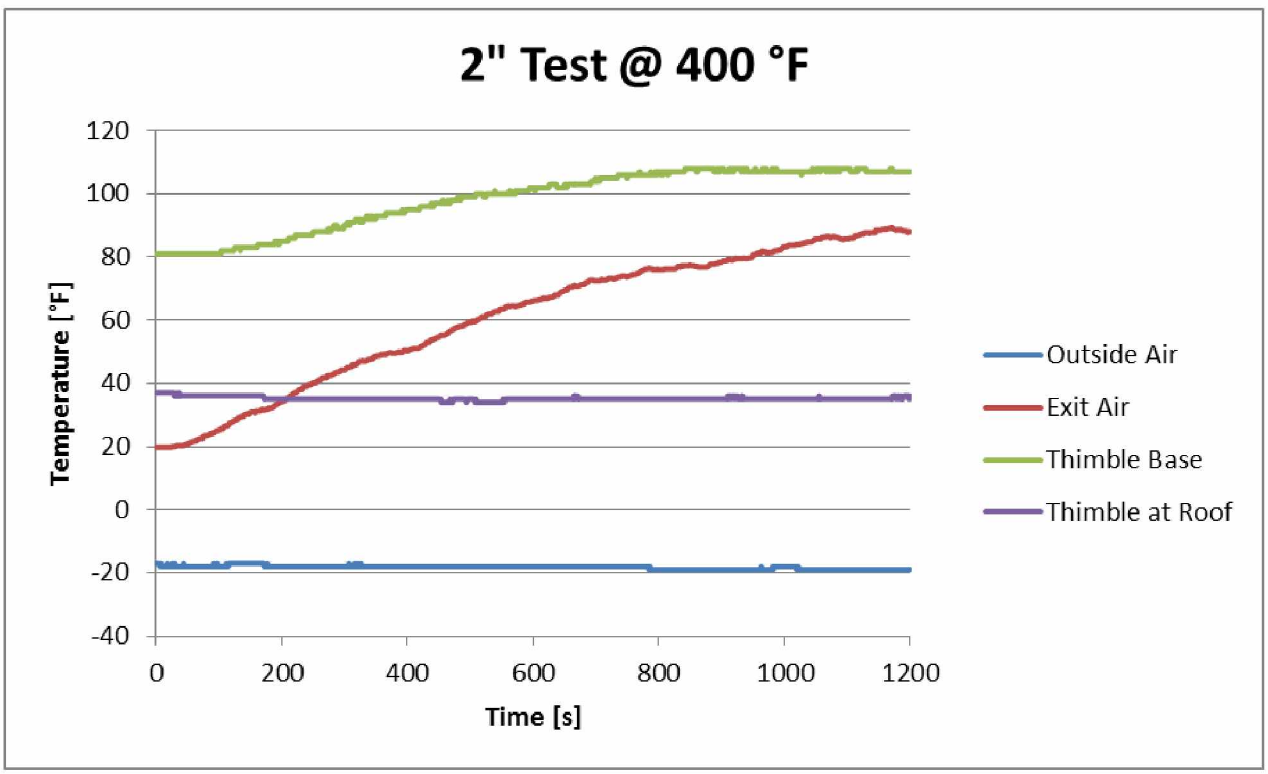

Figure 19: Winter 2" Test at 400°F

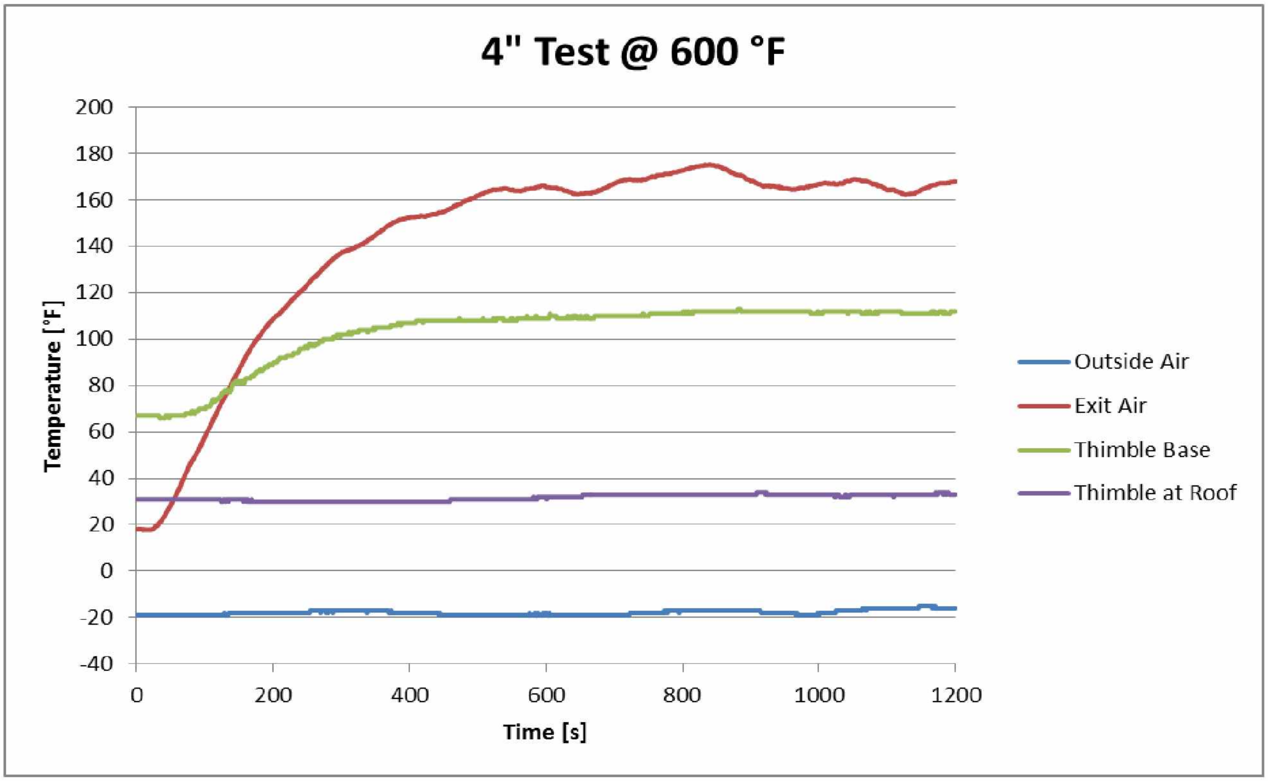

Figure 20: Winter 4" Test at 600°F

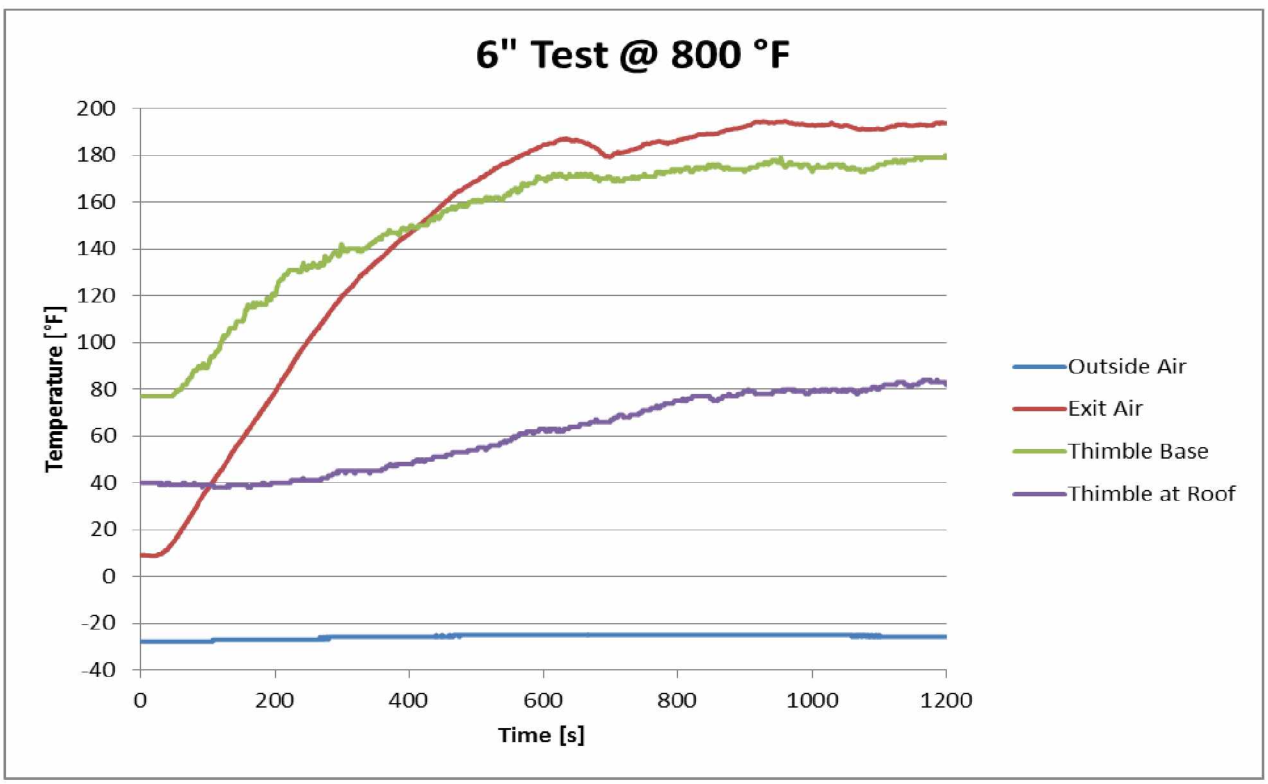

Figure 21: Winter 6" Test at 800°F

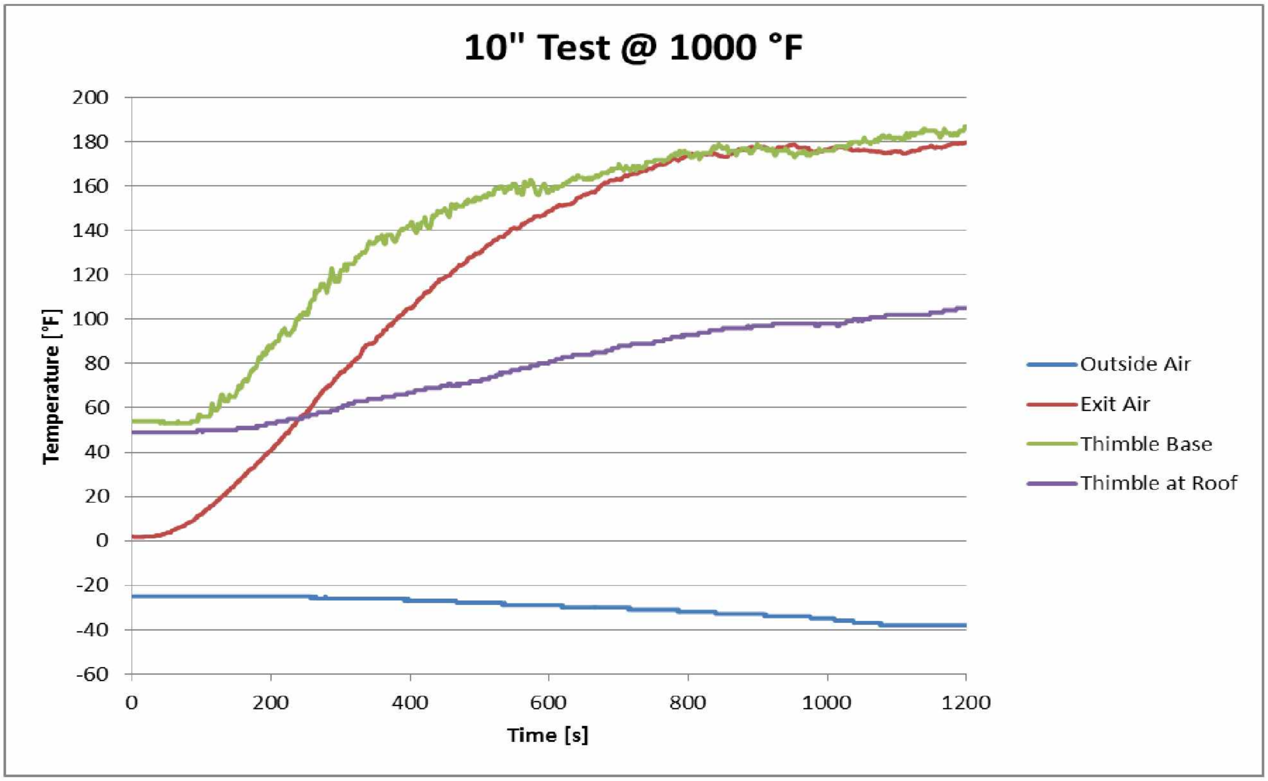

Figure 22: Winter 10" Test at 1000°F

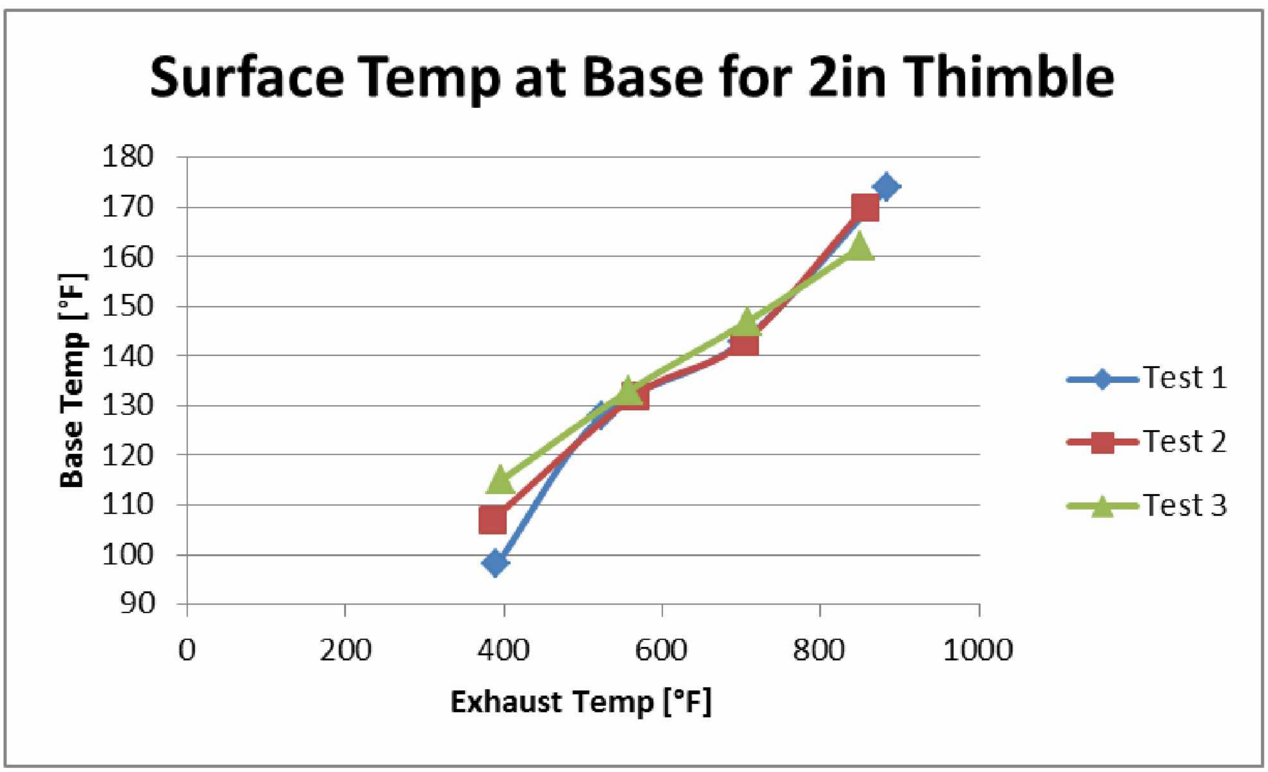

Figure 23: Temperature of the exterior skin at the base

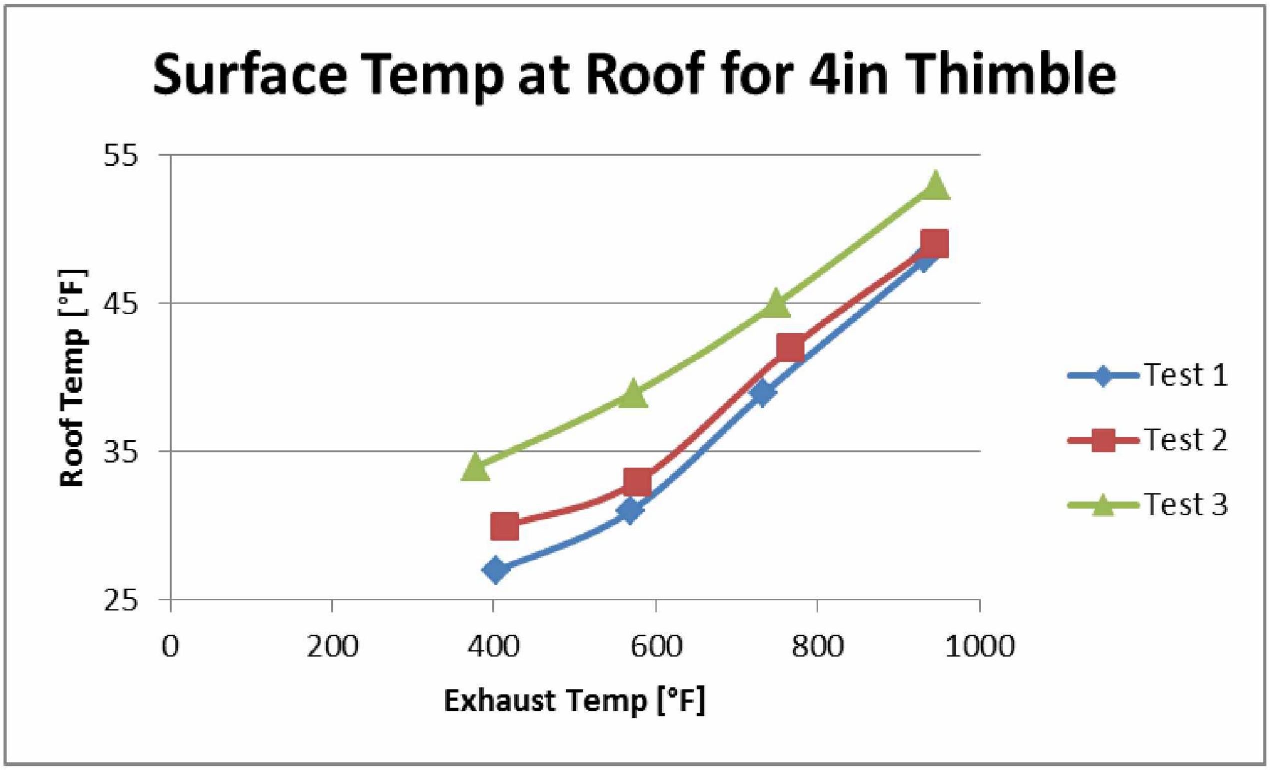

Figure 24: Temperature at the roof material and thimble interface

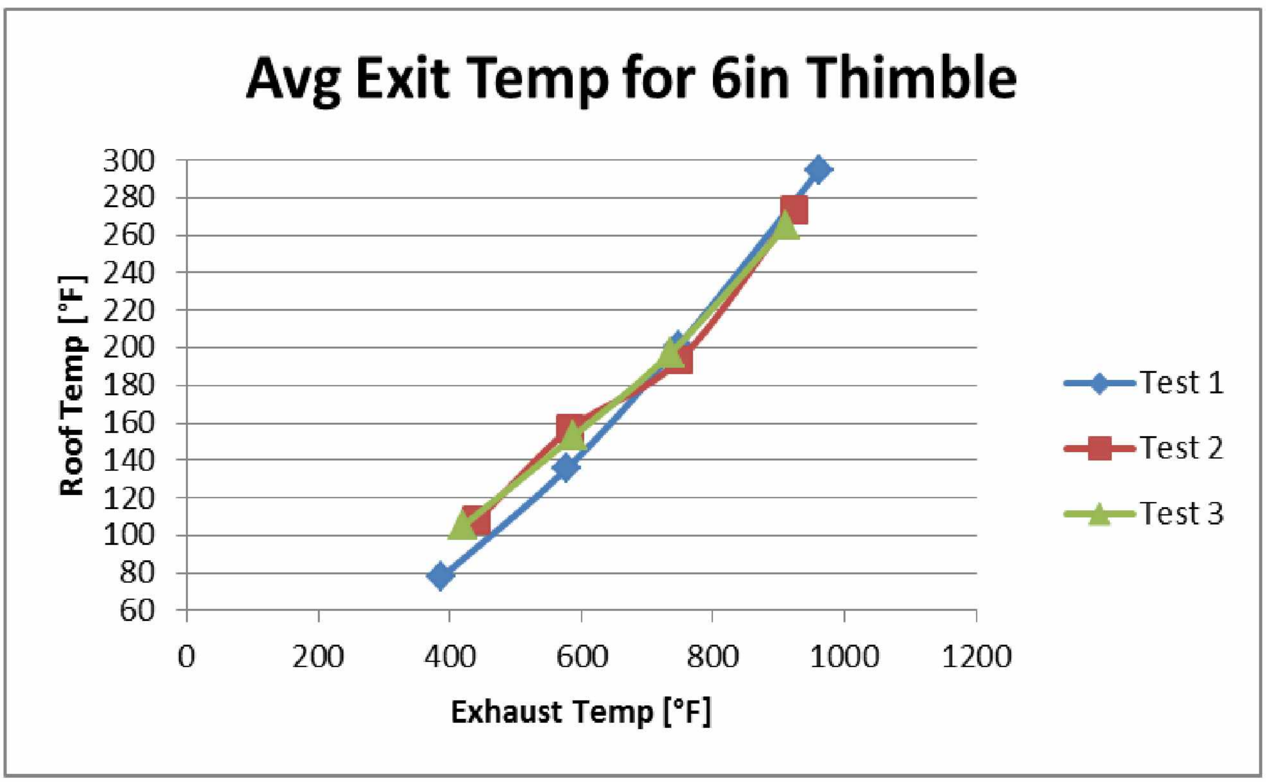

Figure 25: Average exit temperature of cooling air from inner annular ring

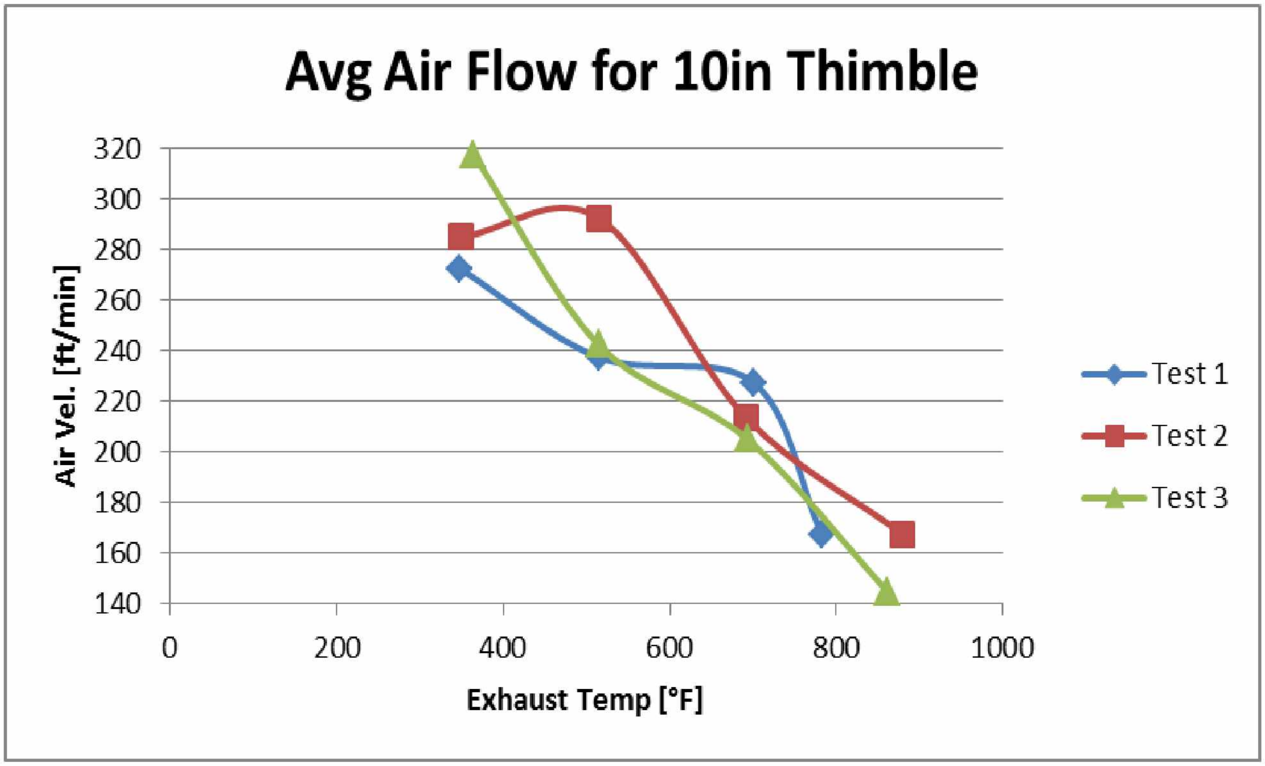

Figure 26: Cooling Air flow approximately in the center of the outer annulus

Figures 20-23 are good examples of the performance of the four thimbles at appropriate temperatures during the winter months. Temperatures at or below -40°F were not achievable at the time of analysis. Therefore, most of the winter testing was conducted at around -20°F. The surface temperatures (Figures 24&25) behaved very similar to those in the summer just at lower temperatures. This may prove to be an issue since these temperatures are generally below the dew point. The exit temperatures of the cooling air seen in Figure 26 are nearly the same and some are slightly higher than those in the summer. This implies that there is a greater heat transfer because of the larger ΔT. These similarities end with the air flow velocity seen in Figure 27 curiously decreasing as the flue gas temperature increases. This may be because as the temperature increases, the radiant heat transfer to the inner annulus wall increases and this creates a great turbulence in the outer annulus due to the buoyant effect of the heated air on the skin and the large temperature differential. Whatever the case, it allows time for more heat to be transferred to the cooling air.

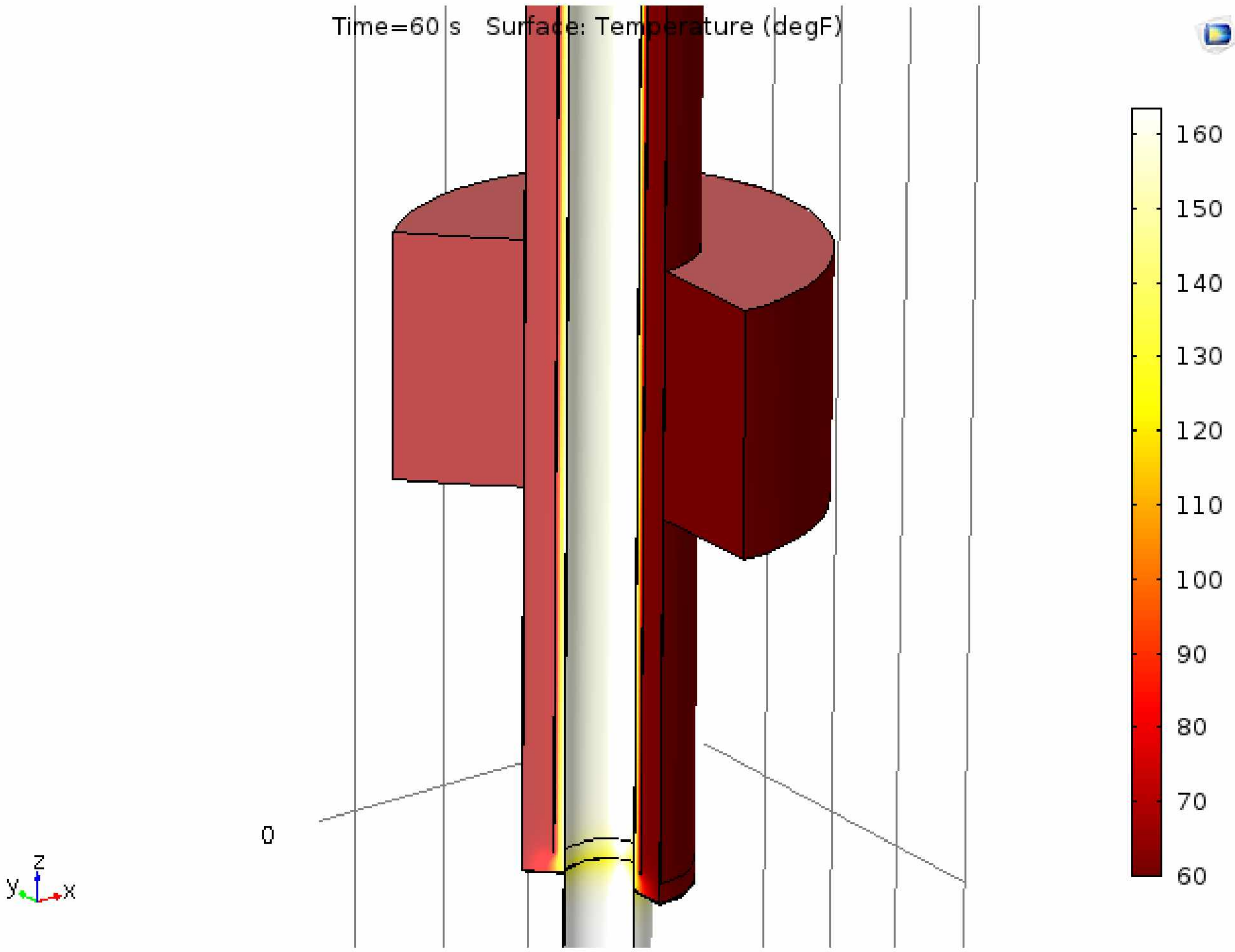

Figure 27: Summer 4" Temperature Gradient

From Figure 28 it can be seen that it is close to 100°F hotter in the inner annulus than in the outer. This supports what was seen above in that the cooling air is pulling off a great deal of heat while maintaining a relatively cool surface temperature at the building-thimble interface. Looking at the base of the model shown in Figure 28 also supports what was seen in the graphs above where the base surface temperature was much hotter than the rest of the outer surface. It can be seen that the heat transfer here is unimpeded by the inner annular ring and can thus have a greater effect on the outer surface temperature.

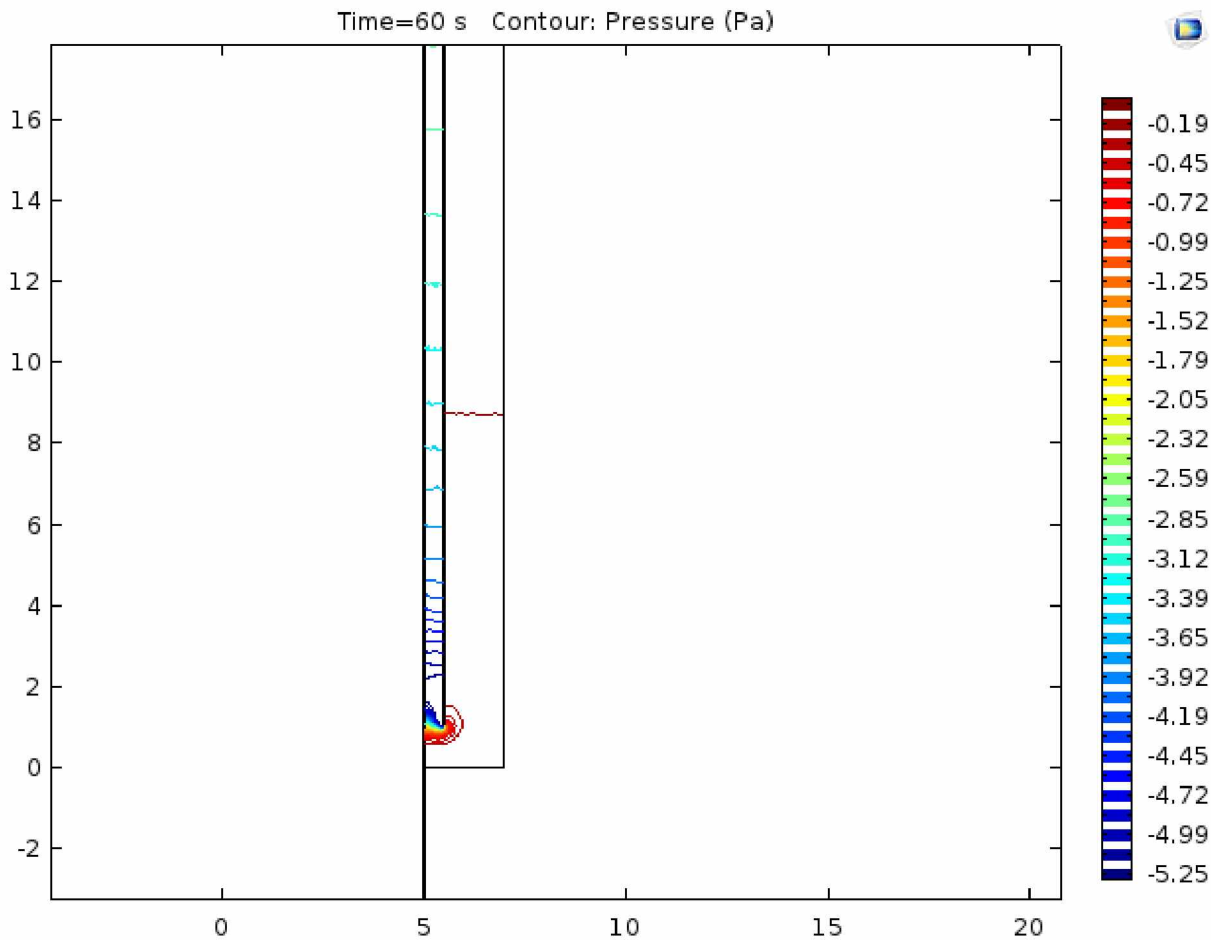

Figure 28: Summer 10" Pressure Gradient

Figure 29 further supports what was observed in Figure 28. The thermal gradient induces a pressure gradient. It can be seen that there is a relatively large pressure differential right at the turning point from the outer annular ring into the inner annular ring. This is a huge driving factor in creating the natural convective process for the thimble to function.

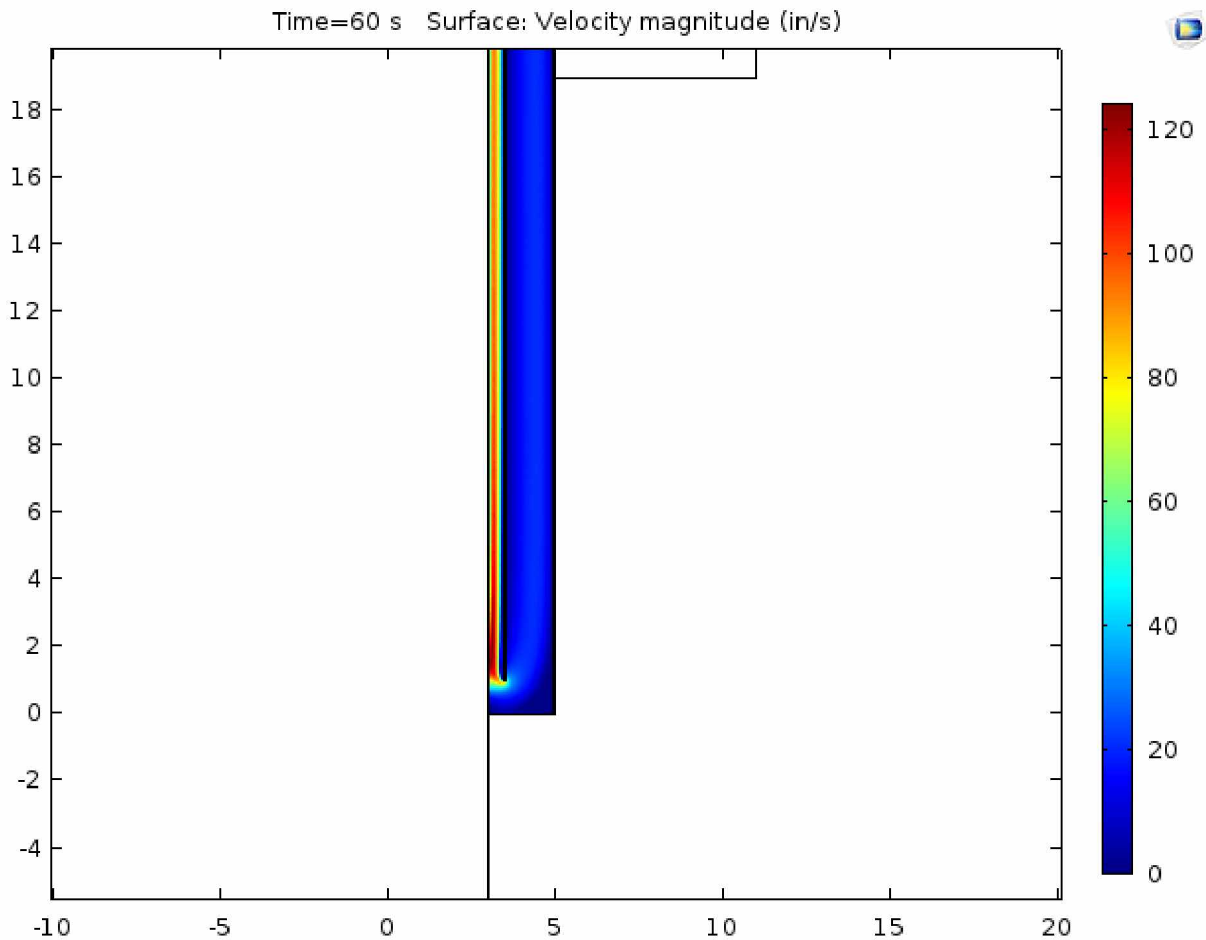

Figure 29: Summer 6" Velocity Gradient

Figure 30 shows the culminating effect of the previous two graphs. That is to say that the thermal gradient causes a pressure gradient that then causes fluid flow. The figure carries attributes from the previous two graphs. Notice the relatively large difference in velocity between the inner and outer annular spaces and the steep velocity gradient at the turning point that is the effect of the mirrored gradient in Figure 29.

Results Discussion:

Appliance type	Exhaust Temperature	
	(°F)	(°C)
Chemical Oxidation	1350 - 1475	730 - 800
Annealing furnace	1,100 - 1,200	590 - 650
Fluidized-bed combustor	1,600 - 1,800	870 - 980
Natural-gas fired heating appliance with draft hood	360	182
Liquefied-petroleum gas-fired heating appliance with draft hood	360	182
Gas-fired heating appliance, no draft hood	460	238
Glass melting furnace	1,200 - 1,600	650 - 870
Oil-fired heating appliance, residential	560	293
Oil-fired heating appliance, forced draft over 400.000 Btu/h	360	182
Conventional incinerator	1,400	760
Controlled air incinerator	1,800 - 2,400	982 - 1,316
Pathological incinerator	1,800 - 2,800	982 - 1,538
Gas turbine exhaust	700 - 1,100	370 - 590
Diesel exhaust	1,000 - 1,200	540 - 650
Ceramic kilns	1,800 - 2,400	982 - 1,316

Figure 30: Common Exhaust Temperatures

The 400°F range is an important range because as can be seen from Figure 27 this is about the bottom end of where most home fuel burning heaters reside. The 600 °F range on the other hand offers results for the top end of most home heating systems. The thermocouples used were unable to gather data above 1200 °F, so 1000°F was the highest range obtained and it can be seen from Figure 31 that this is the bottom end of many industrial processes and therefore lends some insight into the application of this design in the industrial market. 800 °F was also measured to establish an even spread of data for the computer modeling. It proved difficult to maintain the exhaust temperature precisely at these target temperatures. The valve on the propane torch required constant adjustment. This is most likely due to cooling of the propane gases in the tank as it was used which resulted in lowered pressure and flow rate. This could possibly be averted by using a much larger propane tank or fuel source. The stove pipe dimensions were chosen based on common dimensions used here in Alaska. A 2 inch pipe would be typical of a fuel oil burning space heater which is used heavily here in Alaska. A 4 or 6 inch pipe would be typical of a wood stove or boiler system. A 10 inch pipe would be applicable for large diesel gen-sets.

In collecting the flow data it was observed that the velocities were highest at the outer edge of the outer annulus and that they dropped off rapidly as the anemometer probe moved towards the center. In fact if the tip is pushed in all the way to the inner annulus, it generally read zero. Forced air/wind had a significant effect on performance as well, often times causing the flow rates to spike 3-5 times their normal levels resulting in much cooler surface temperatures. From the data it is seen that during summer conditions there is a definite increase in air flow with increase in exhaust temperature as expected. When the air is cooler however, the flow rate decreases and the current COMSOL model may not be adequately capturing the radiant heat transfer effects, because radiant heat transfer physics were not used, between the flue ring and the inner annular ring. This is most likely causing a buoyant effect in the outer annular ring that is acting against the induced draft. By examining the results section and graphs in the appendix it can be seen that although during winter conditions the flow rates start out higher, as the working temperature reaches 1000°F the flow rates begin to converge with those of the summer conditions. It could be hypothesized that this is due to radiant heat transfer playing a greater role as mentioned above. With the exception of a few fully occluded tests, the surface temperature at the envelope was kept well below the UL listed combustion temperature.

The work done in COMSOL very closely mirrored the results from experiment. At first a constant wall temperature was used to simulate the heat source, but this resulted in excessively high exit temperatures for the cooling air. Following that a heat flux solution was used and a heat transfer coefficient was implemented. To determine this heat transfer coefficient, initially calculations were done and a value of approximately 9 $\left[\frac{lb_m}{s^3 \cdot {}^\circ\mathrm{F}}\right]$(see Appendix). This proved to be too low for the COMSOL model and a value of 50 $\left[\frac{lb_m}{s^3 \cdot {}^\circ\mathrm{F}}\right]$ produced more applicable results. This seems to imply that more variables are not being taken into account in the existing COMSOL model.

Conclusion: Further models might incorporate varying width annular spacing to affect the cooling air velocities. This might help to eliminate hot and cold spots e.g. the below dew point temps and hot base temps. Adding this variance would add additional production costs, so maybe adding some measure of insulation to the outer layer might eliminate these issues and still keep the thimble cost reasonable. Testing at much higher temperatures to further define the trend of air flow velocities would also be recommended.

From the data it can be seen that the design is quite viable and performs fully as expected. The skin temperature at the roof stays well below the 160°F threshold in most cases and only crosses it for stack temperatures of 800°F and 1000°F when the inlet is at least two-thirds occluded (See Appendix). This is good news for the design; since in the case of runaway burners, the thimble will still perform given that it is kept clean of debris. From here the data will be further compared against more refined computer simulations and used to increase the accuracy of those simulations. With further research and perseverance, the optimum dimensions for these thimbles should be discovered and will most likely perform better than the current ones and quite possibly eliminate the few high and low temperature issues. This research seems to suggest that one day this product will replace current thimble designs.

Appendix

Sample Calculations for 2” Thimble at 400 °F

$$T_{A,Hot} = 370\ [°F]$$

$$T_{A,Cold} = 70\ [°F]$$

$$T_{B,Hot} = 307\ [°F]$$

$$T_{B,Cold} = 125.5\ [°F]$$

$$\Delta T_A = 300\ [°F]$$

$$\Delta T_B = 182\ [°F]$$

$$D_A = 2\ [in]$$

$$D_B = 3\ [in]$$

$$D_C = 6\ [in]$$

$$V_{Avg} = 100\ \left[\frac{ft}{min}\right] = 6{,}000\ \left[\frac{ft}{hr}\right]$$

$$\rho_{Air@70°F} = 0.0749\ \left[\frac{lb_m}{ft^3}\right]$$

$$Cp_{Air@70°F} = 0.241\ \left[\frac{BTU}{lb_m \cdot °F}\right]$$

$$A_{Cross-Sectional} = \frac{\pi}{4}\left({D_C}^2 - {D_B}^2\right) = \frac{\pi}{4}(6^2 - 3^2)[in^2] = 21.21\ [in^2] = 0.147\ [ft^2]$$

$$\dot{Q} = V_{Avg} \cdot A_{CS} \cdot \rho_{Air@70°F} \cdot Cp_{Air@70°F} \cdot \left(T_{B,Cold} - T_{A,Cold}\right)$$

$$= (6{,}000)\left[\frac{ft}{hr}\right](0.147)[ft^2](0.0749)\left[\frac{lb_m}{ft^3}\right](0.241)\left[\frac{BTU}{lb_m \cdot °F}\right](125.5 - 70)[°F]$$

$$= 885.4\left[\frac{BTU}{hr}\right]$$

$$A_S = \pi D_A H = \pi(2)[in](65)[in] = 408.4\ [in^2] = 2.84\ [ft^2]$$

$$LMTD = \frac{\Delta T_A - \Delta T_B}{\ln\frac{\Delta T_A}{\Delta T_B}} = \frac{(300 - 182)[°F]}{\ln\frac{300}{182}} = 235.8\ [°F]$$

$$h = \frac{\dot{Q}}{A_S \cdot LMTD} = \frac{(885.4)\left[\frac{BTU}{hr}\right]}{(2.84)[ft^2](235.8)[°F]} = 1.32\ \left[\frac{BTU}{hr \cdot ft^2 \cdot °F}\right] = 9.21\ \left[\frac{lb_m}{s^3 \cdot °F}\right]$$

Heat Transfer Coefficient Calculation Results for other Thimble Sizes at 400°F

$$h_{4in} = 10.4 \left[\frac{lb_m}{s^3 \cdot \text{°F}}\right]$$

$$h_{6in} = 8.1 \left[\frac{lb_m}{s^3 \cdot \text{°F}}\right]$$

$$h_{10in} = 9.5 \left[\frac{lb_m}{s^3 \cdot \text{°F}}\right]$$

2in Thimble Summer Results

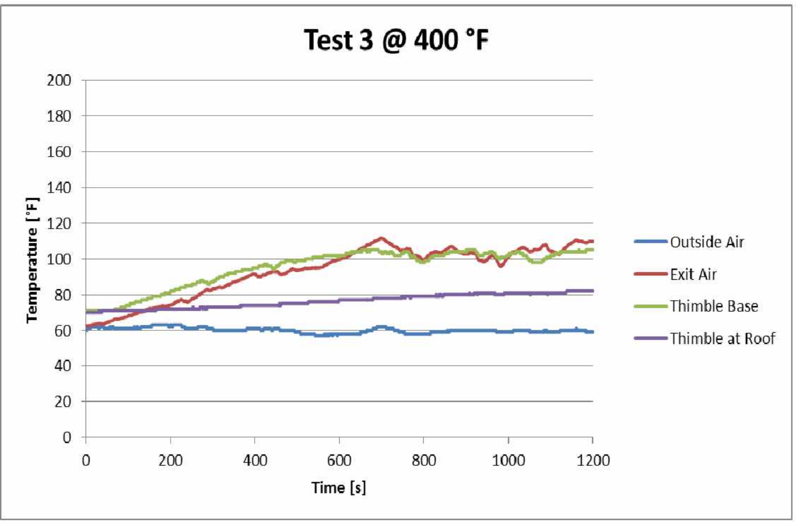

Figure 31: Cool Air Inlet Velocity was Approx. 90 ft/min

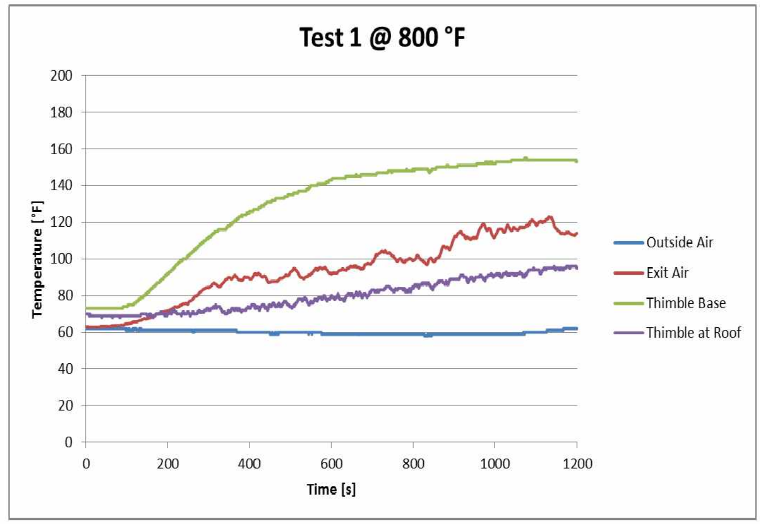

Figure 33: Cool Air Inlet Velocity was Approx. 150 ft/min

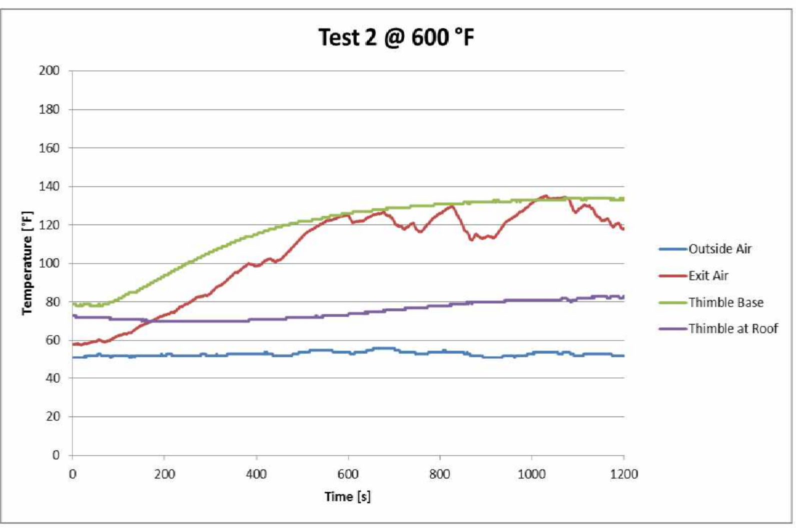

Figure 32: Cool Air Inlet Velocity was Approx. 130 ft/min

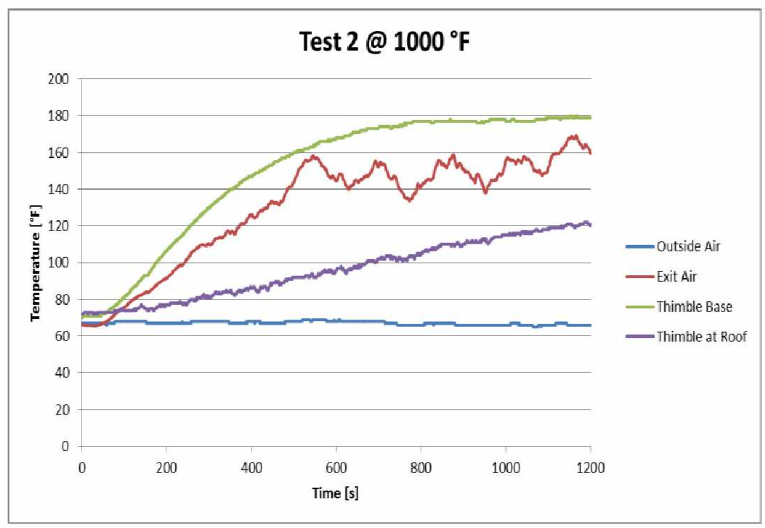

Figure 34: Cool Air Inlet Velocity was Approx. 170 ft/min

2in Thimble Summer Results

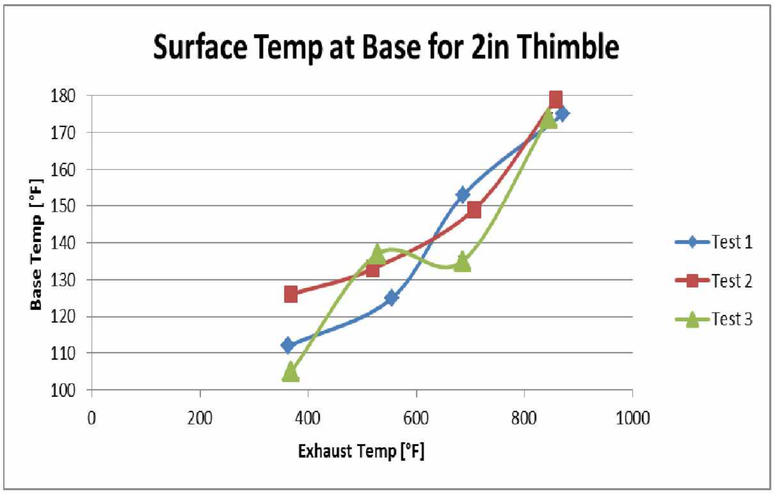

Figure 35: Surface Temperature at the Base of the Thimble

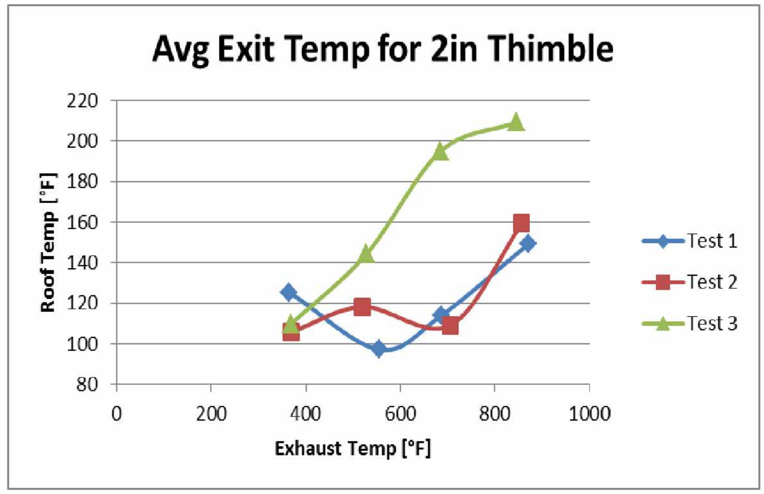

Figure 37: Exit Temperature of Cooling Air

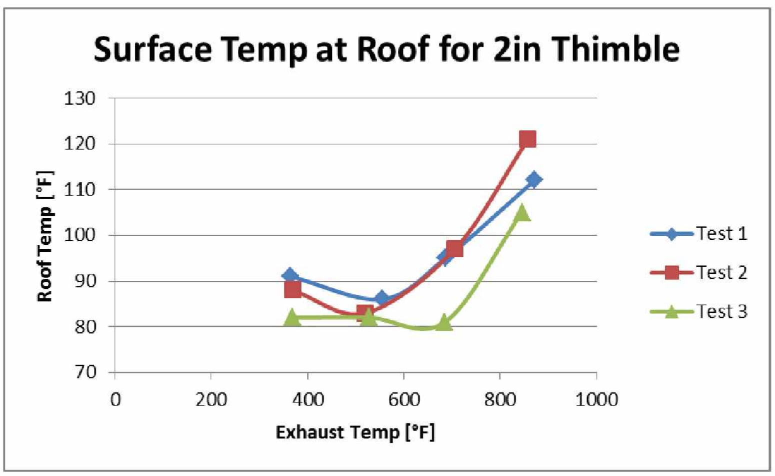

Figure 36: Surface Temperature Contacting Combustible Material

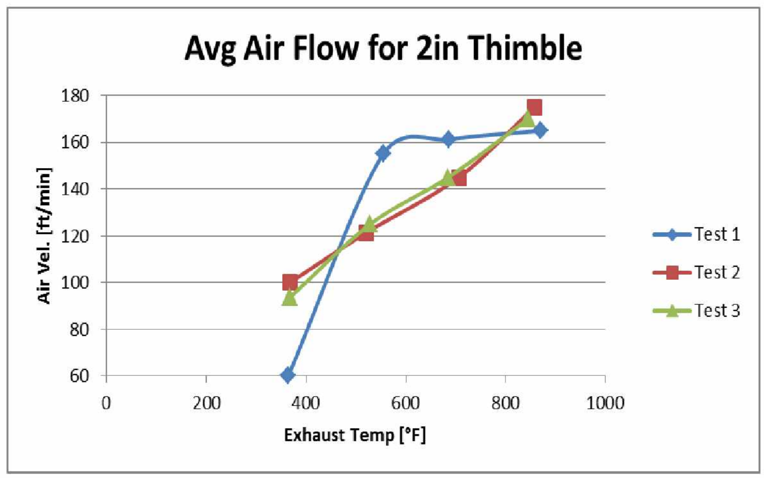

Figure 38: Cooling Air Inlet Velocity

2in Thimble Winter Results

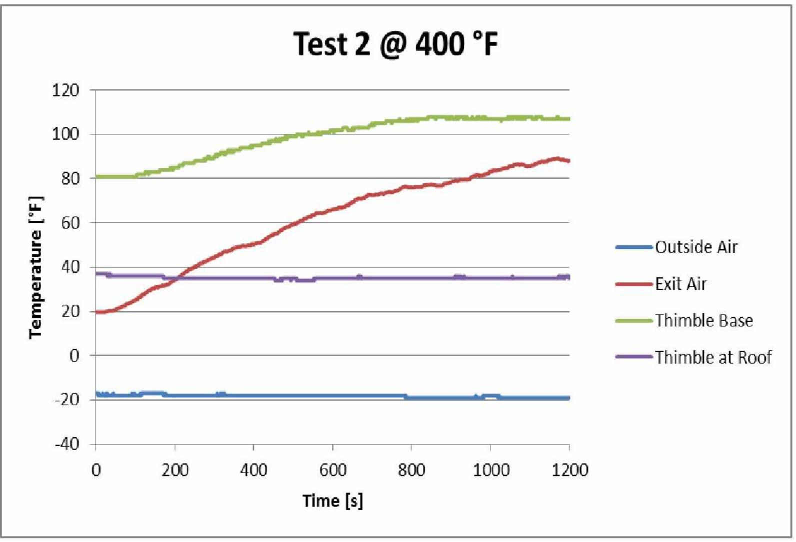

Figure 39: Cool Air Inlet Velocity was Approx. 250 ft/min

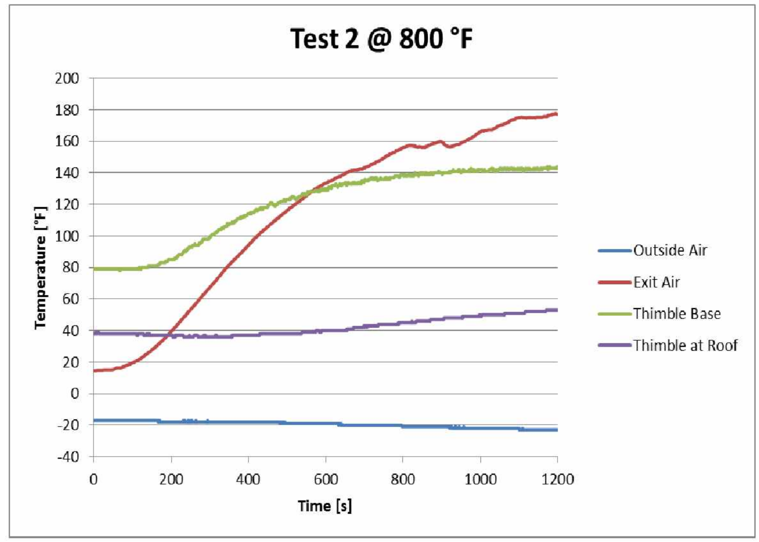

Figure 41: Cool Air Inlet Velocity was Approx. 210 ft/min

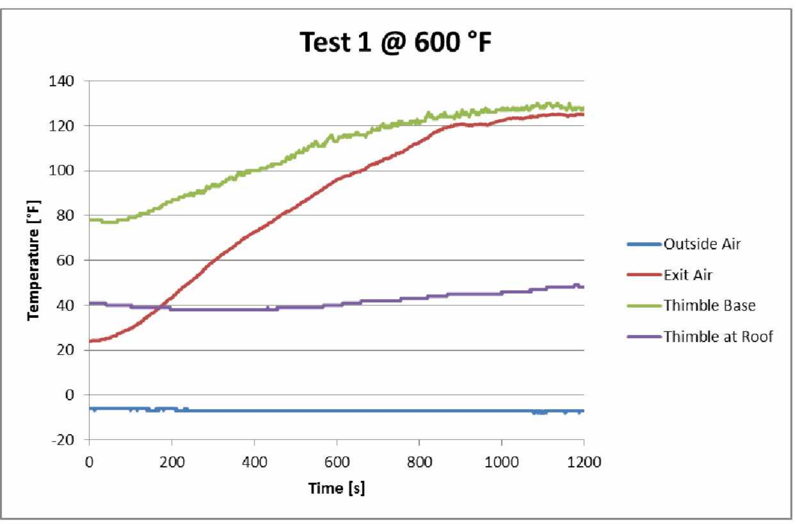

Figure 40: Cool Air Inlet Velocity was Approx. 230 ft/min

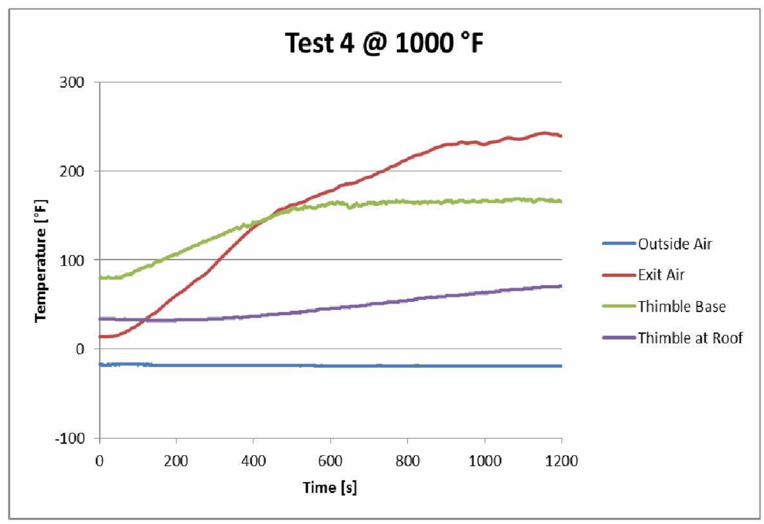

Figure 42: Cool Air Inlet Velocity was Approx. 200 ft/min

2in Thimble Winter Results

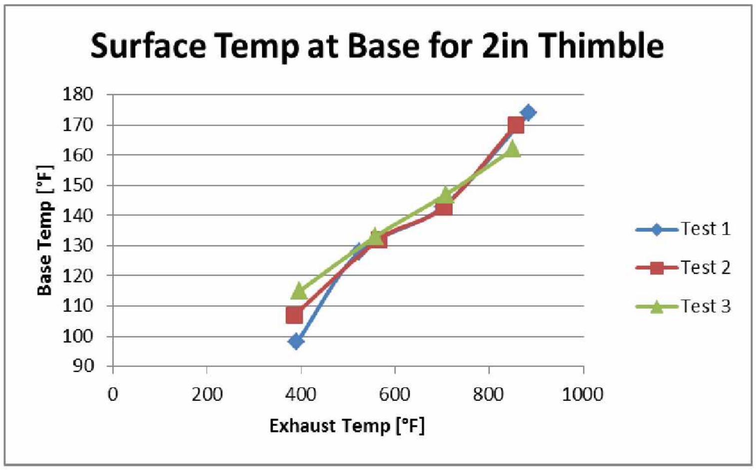

Figure 43: Surface Temperature at the Base of the Thimble

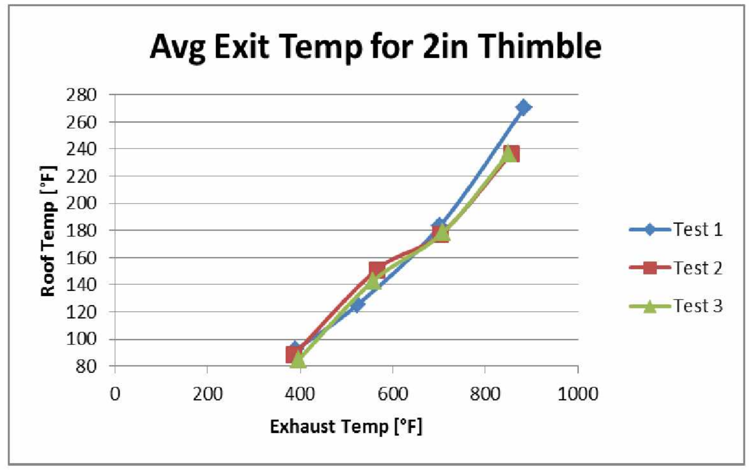

Figure 45: Exit Temperature of Cooling Air

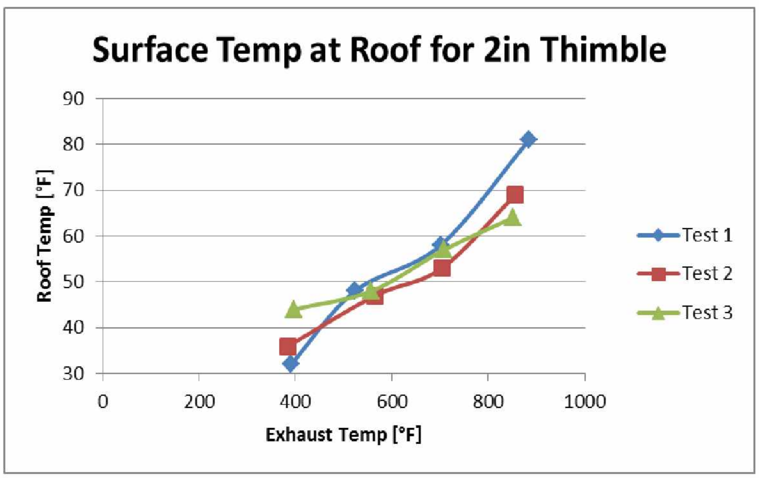

Figure 44: Surface Temperature Contacting Combustible Material

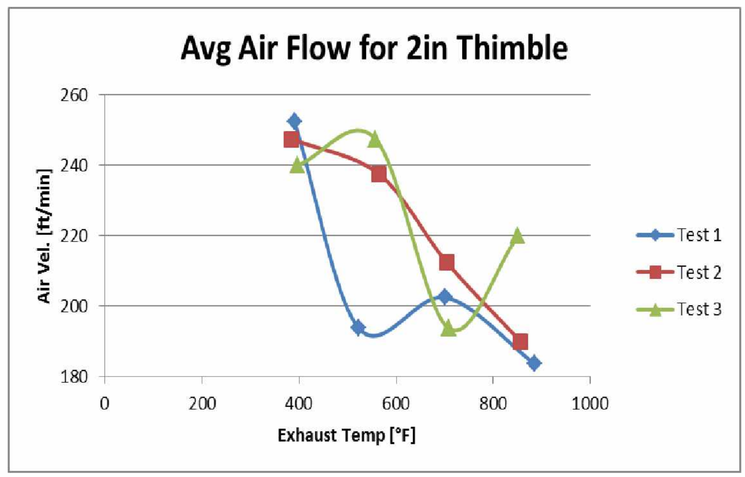

Figure 46: Cooling Air Inlet Velocity

4in Thimble Summer Results

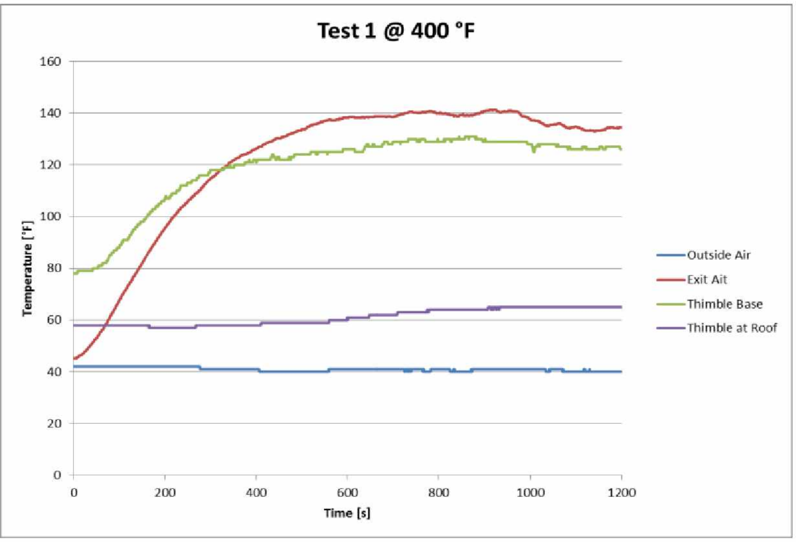

Figure 47: Cool Air Inlet Velocity was Approx. 125 ft/min

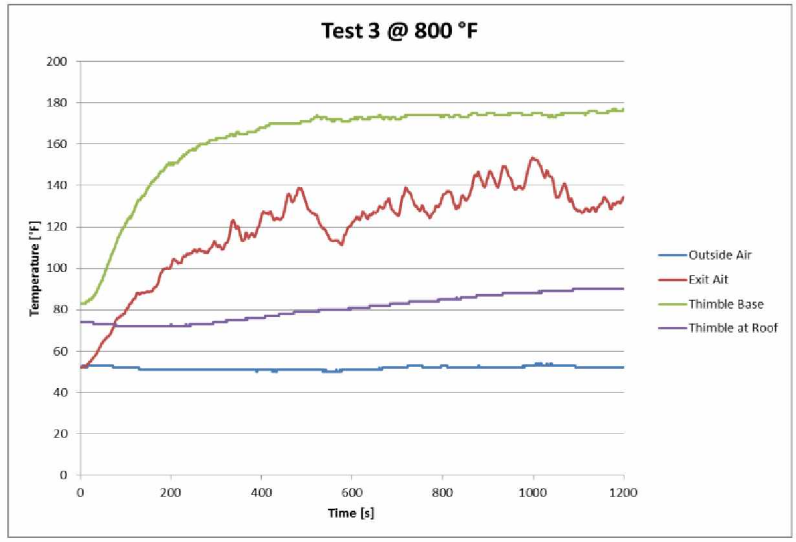

Figure 49: Cool Air Inlet Velocity was Approx. 175 ft/min

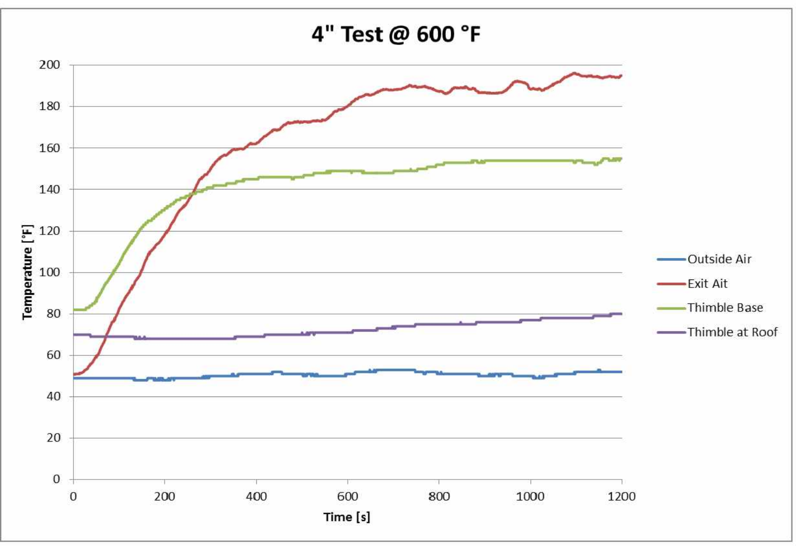

Figure 48: Cool Air Inlet Velocity was Approx. 145 ft/min

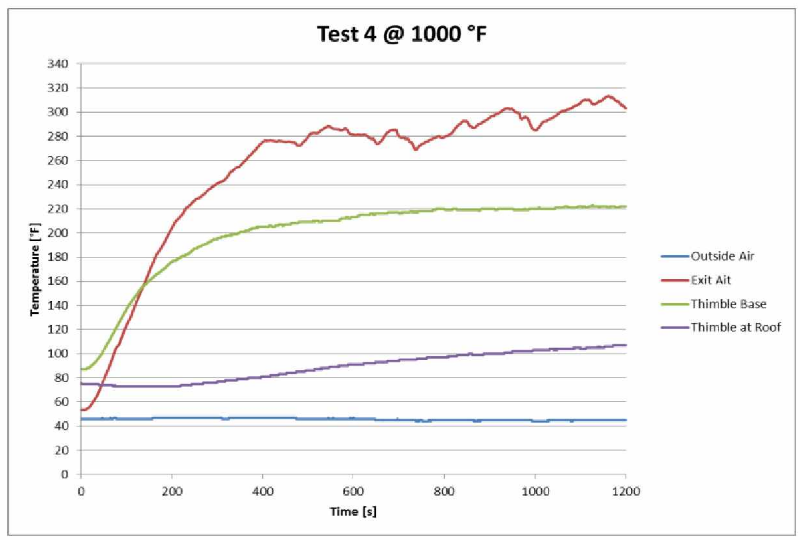

Figure 50: Cool Air Inlet Velocity was Approx. 215 ft/min

4in Thimble Summer Results

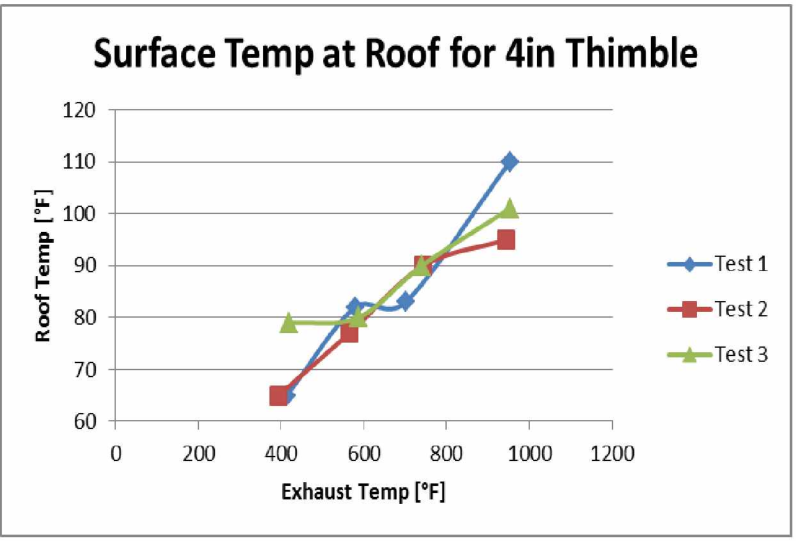

Figure 51: Surface Temperature at the Base of the Thimble

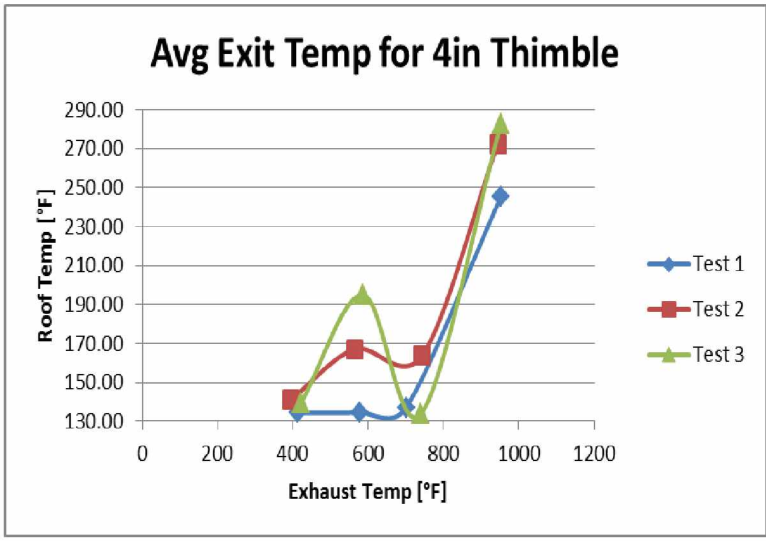

Figure 53: Exit Temperature of Cooling Air

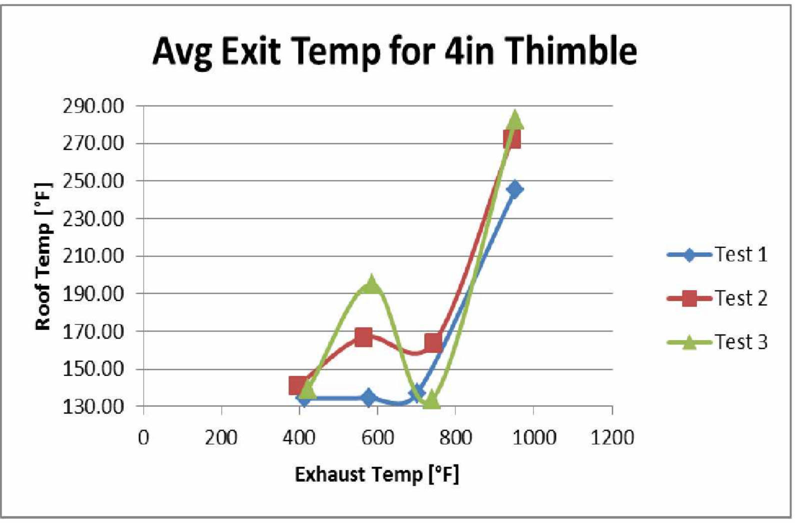

Figure 52: Surface Temperature Contacting Combustible Material

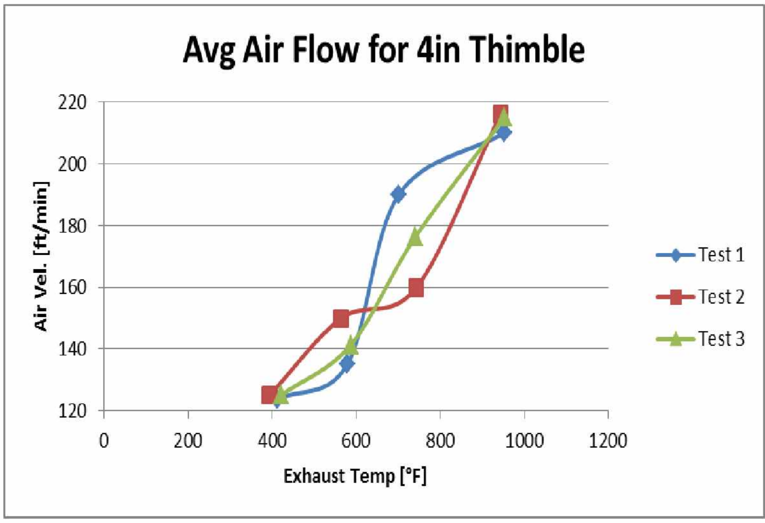

Figure 54: Cooling Air Inlet Velocity

4in Thimble Winter Results

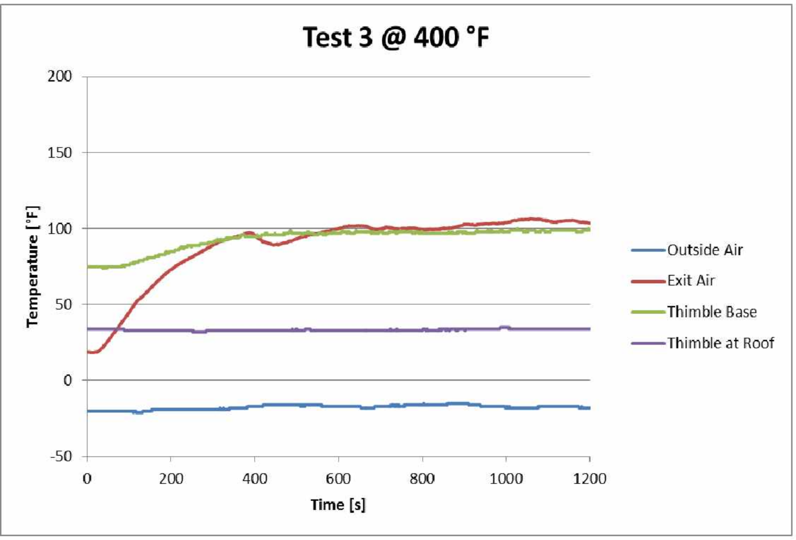

Figure 55: Cool Air Inlet Velocity was Approx. 190 ft/min

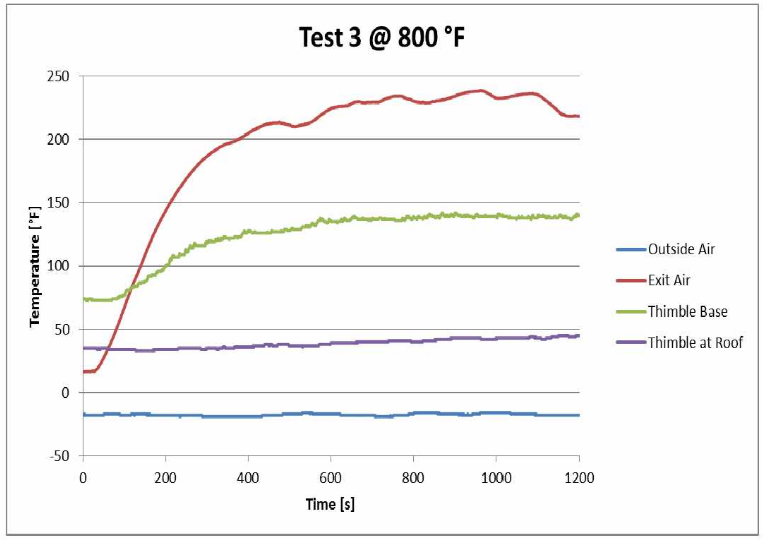

Figure 57: Cool Air Inlet Velocity was Approx. 160 ft/min

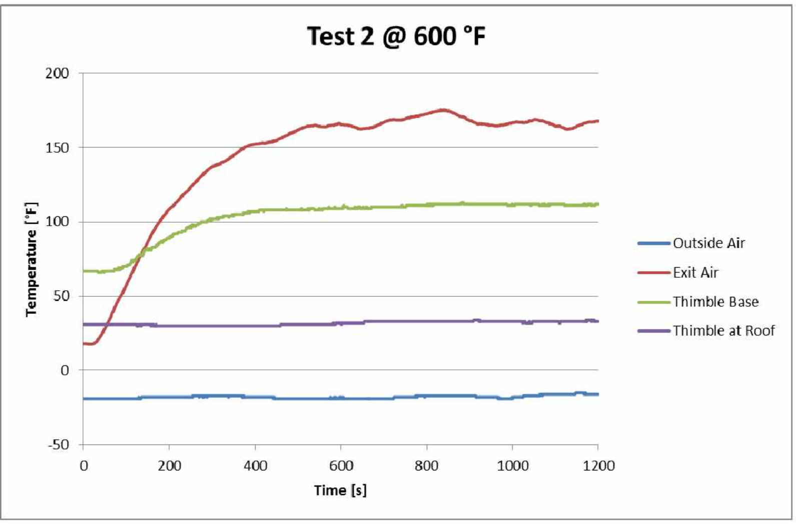

Figure 56: Cool Air Inlet Velocity was Approx. 180 ft/min

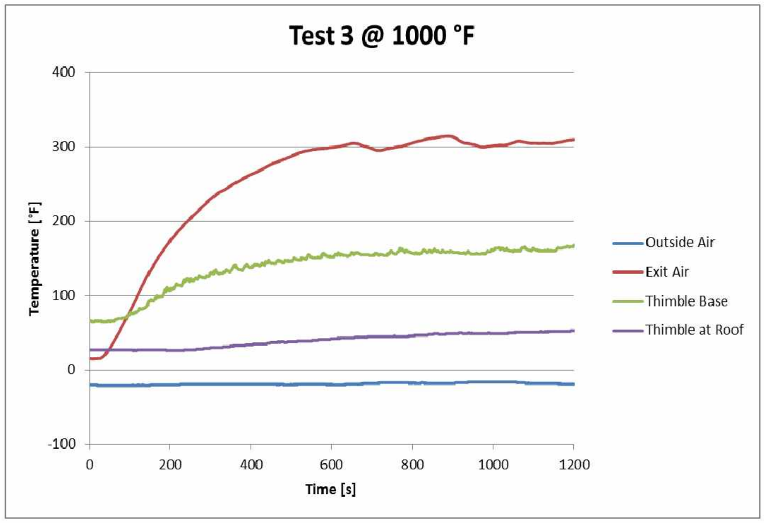

Figure 58: Cool Air Inlet Velocity was Approx. 140 ft/min

4in Thimble Winter Results

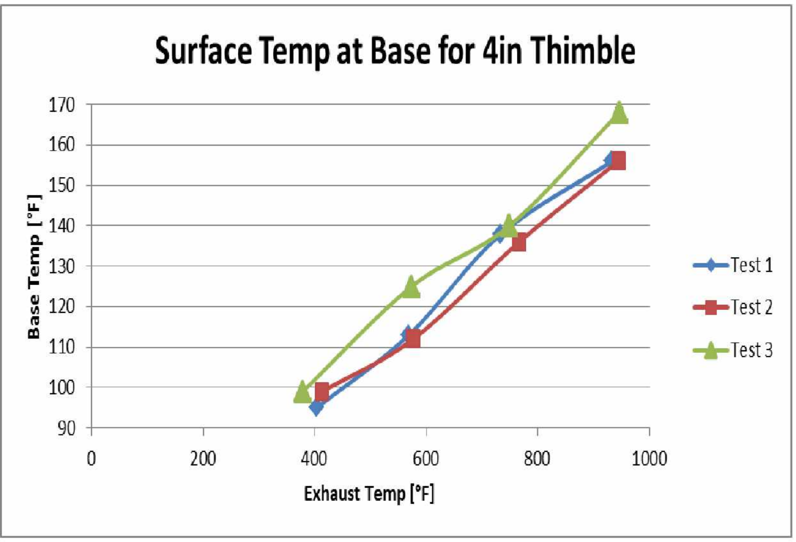

Figure 59: Surface Temperature at the Base of the Thimble

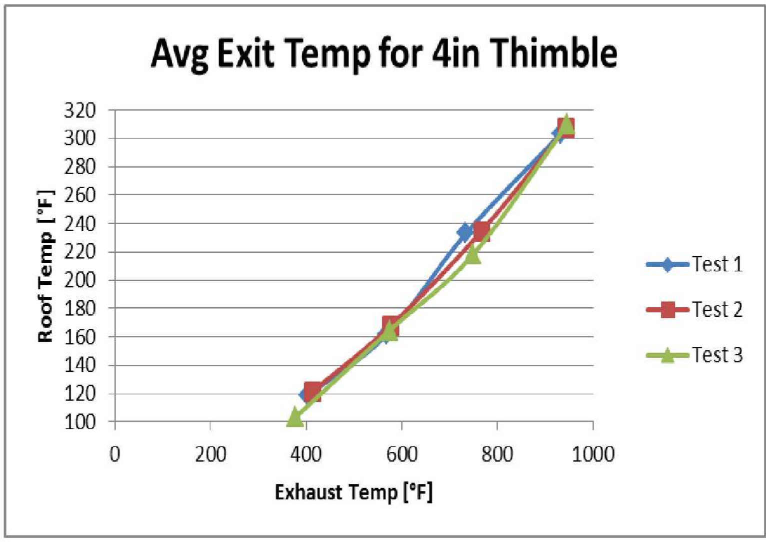

Figure 61: Exit Temperature of Cooling Air

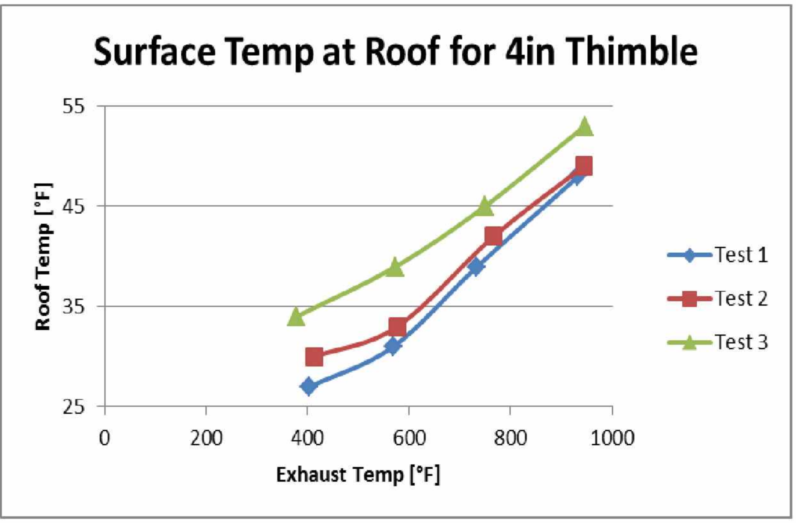

Figure 60: Surface Temperature Contacting Combustible Material

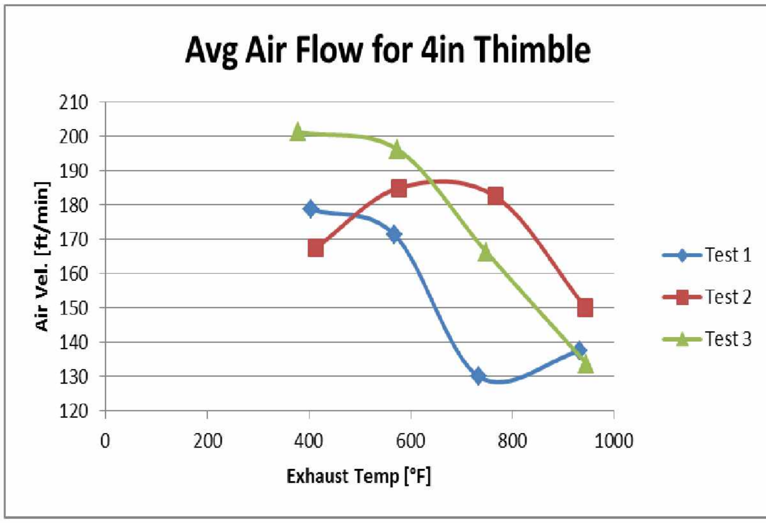

Figure 62: Cooling Air Inlet Velocity

6in Thimble Summer Results

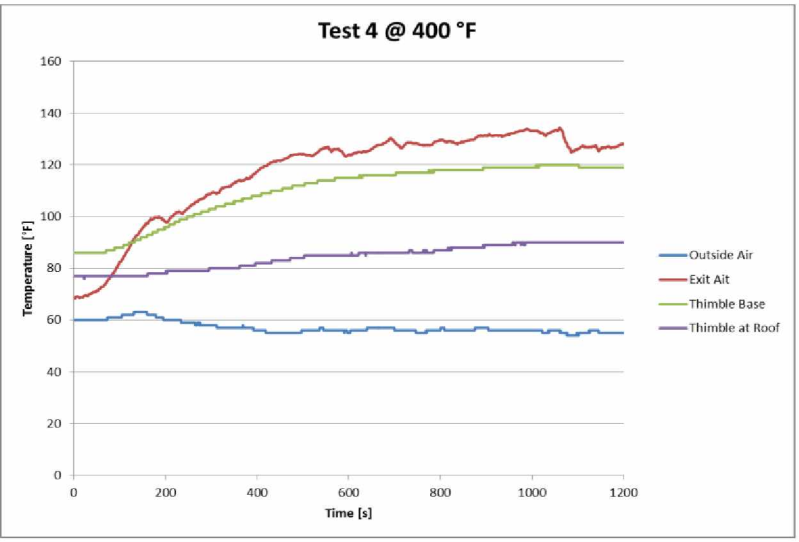

Figure 63: Cool Air Inlet Velocity was Approx. 95 ft/min

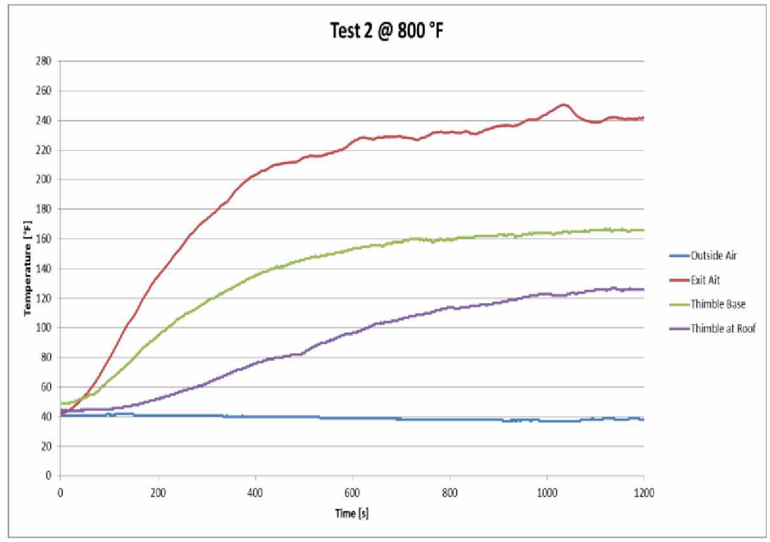

Figure 65: Cool Air Inlet Velocity was Approx. 160 ft/min

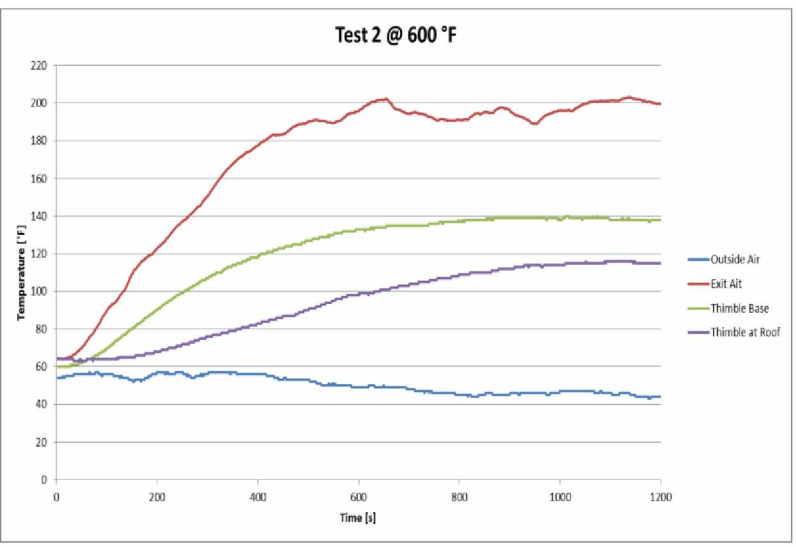

Figure 64: Cool Air Inlet Velocity was Approx. 130 ft/min

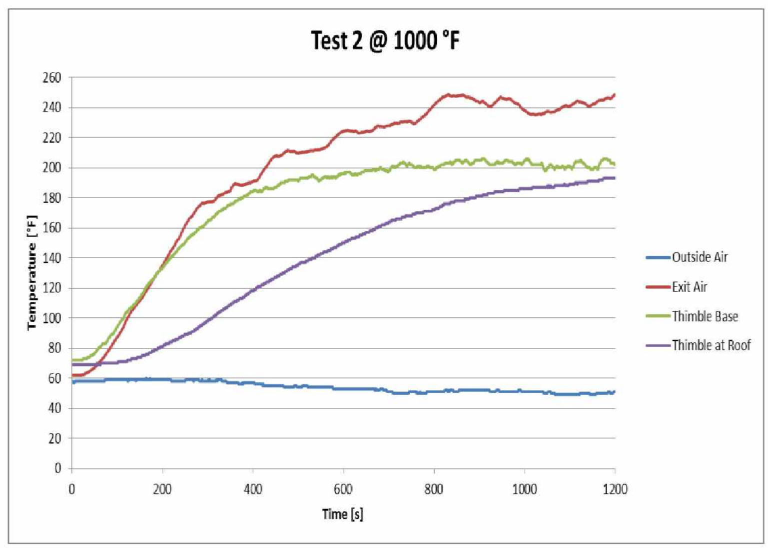

Figure 66: Cool Air Inlet Velocity was Approx. 170 ft/min

6in Thimble Summer Results

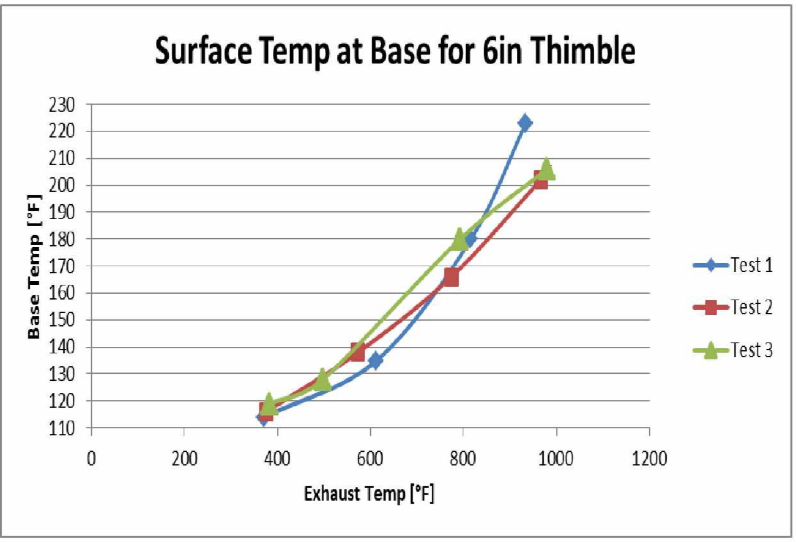

Figure 67: Surface Temperature at the Base of the Thimble

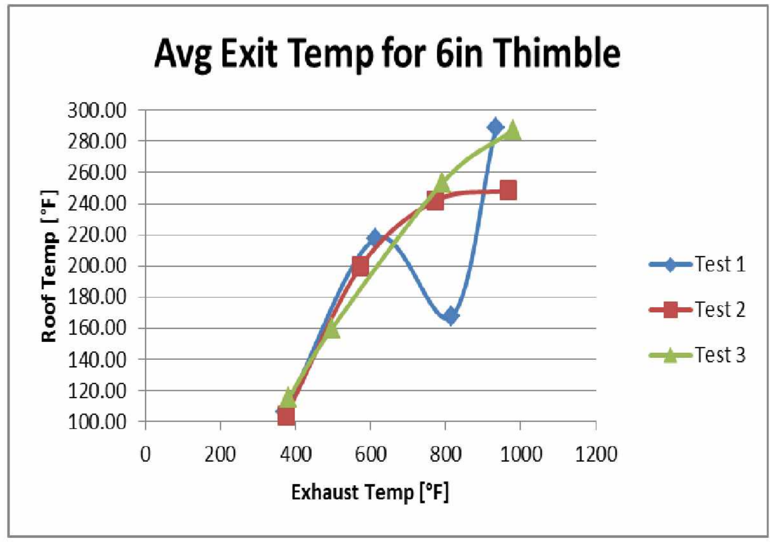

Figure 69: Exit Temperature of Cooling Air

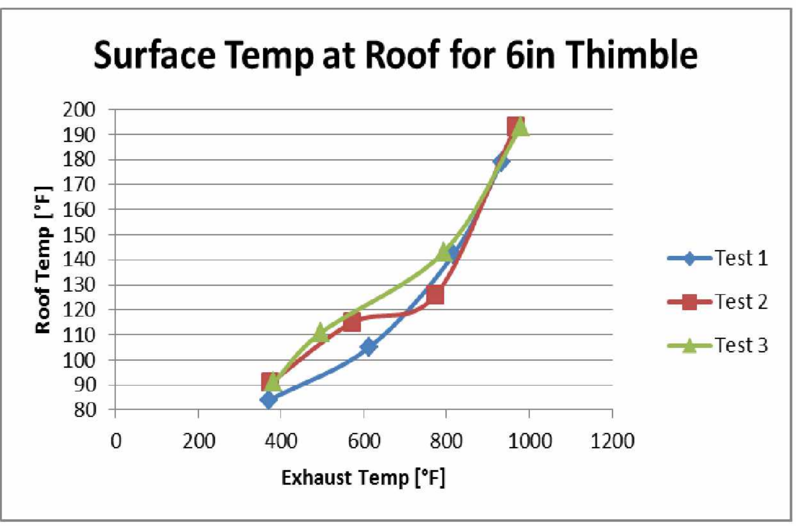

Figure 68: Surface Temperature Contacting Combustible Material

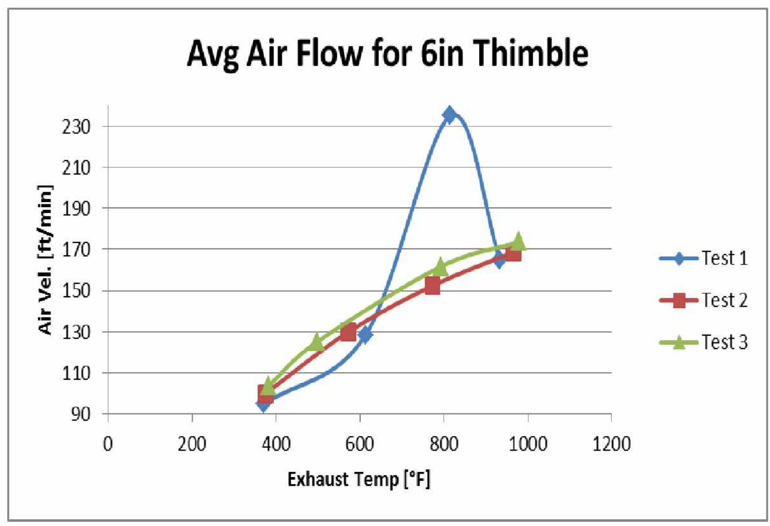

Figure 70: Cooling Air Inlet Velocity

6in Thimble Winter Results

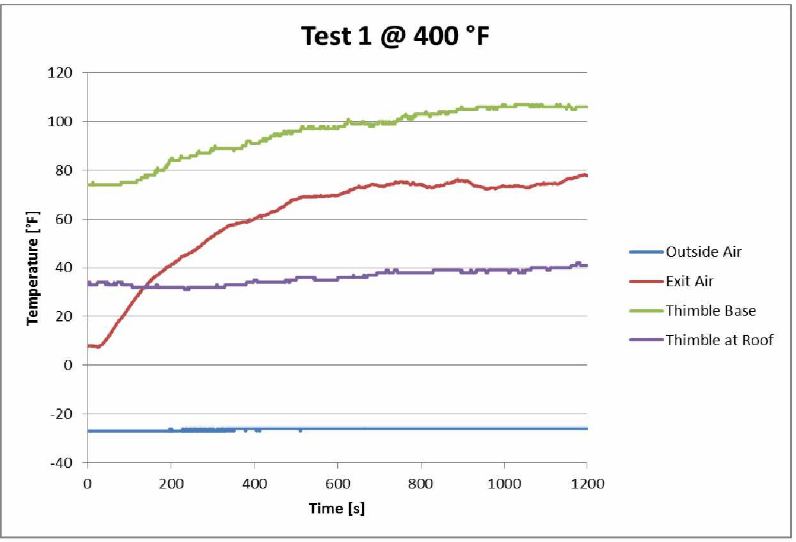

Figure 71: Cool Air Inlet Velocity was Approx. 230 ft/min

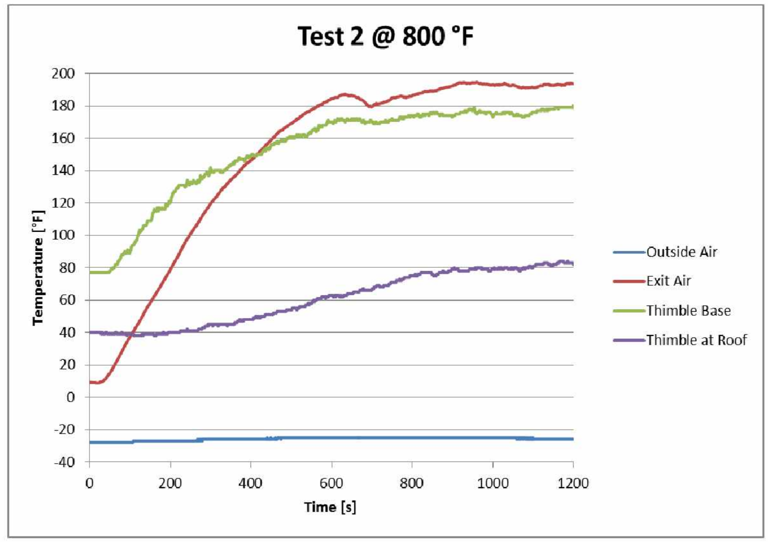

Figure 73: Cool Air Inlet Velocity was Approx. 140 ft/min

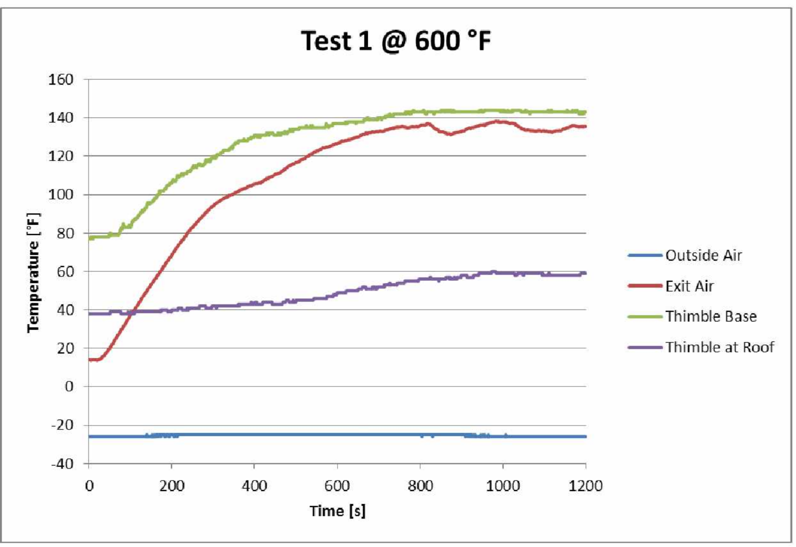

Figure 72: Cool Air Inlet Velocity was Approx. 190 ft/min

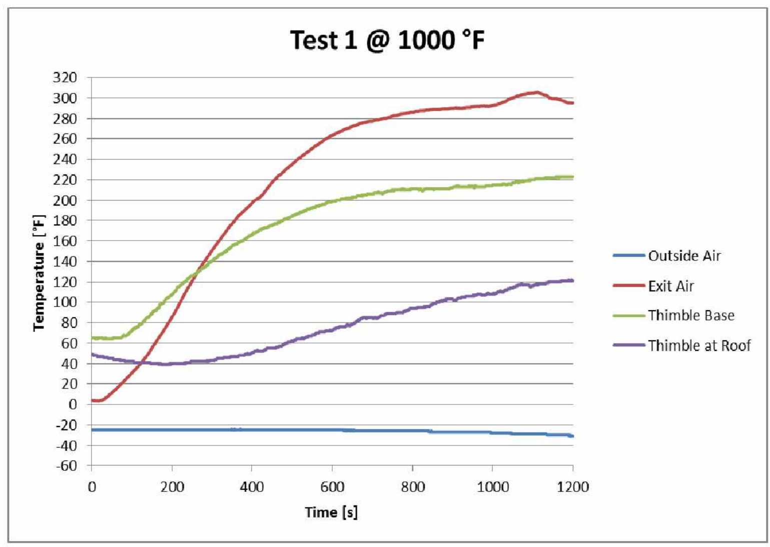

Figure 74: Cool Air Inlet Velocity was Approx. 120 ft/min

6in Thimble Winter Results

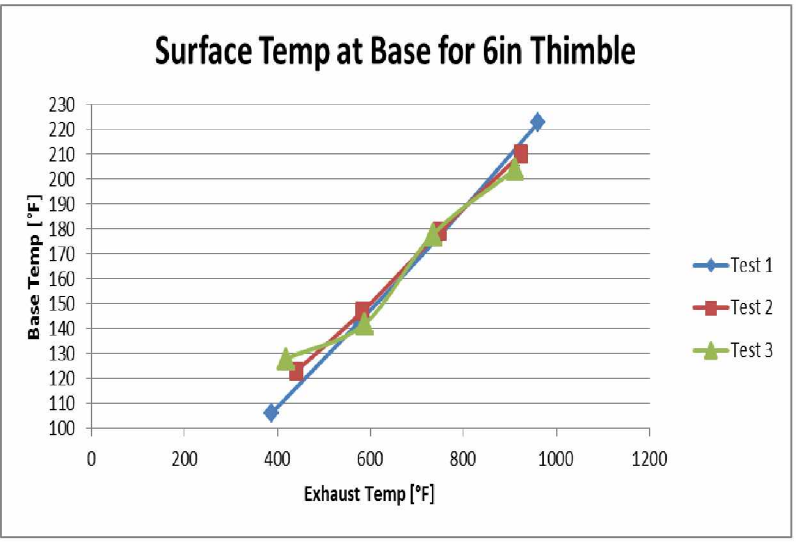

Figure 75: Surface Temperature at the Base of the Thimble

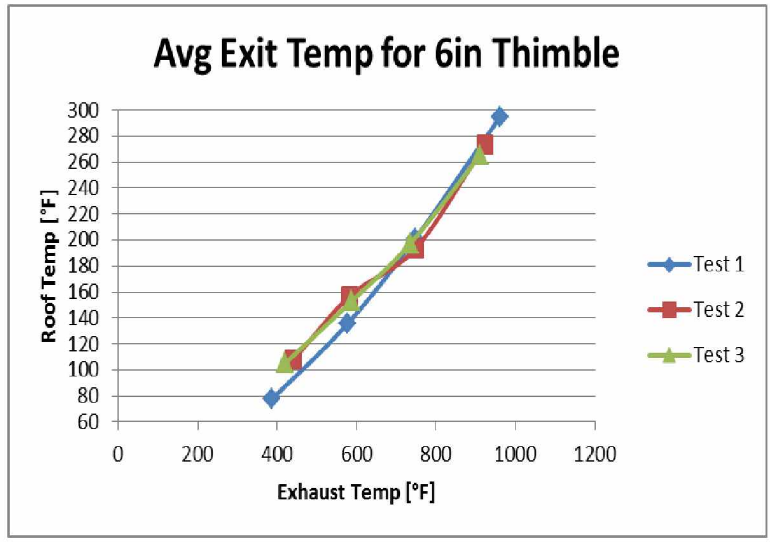

Figure 77: Exit Temperature of Cooling Air

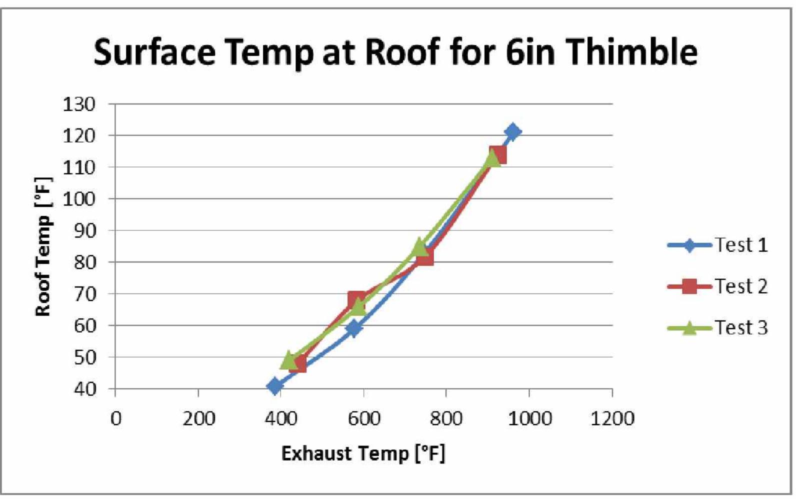

Figure 76: Surface Temperature Contacting Combustible Material

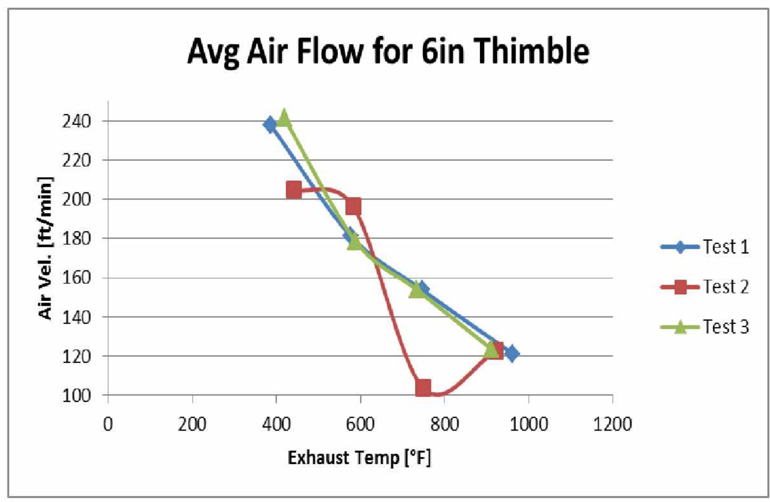

Figure 78: Cooling Air Inlet Velocity

10in Thimble Summer Results

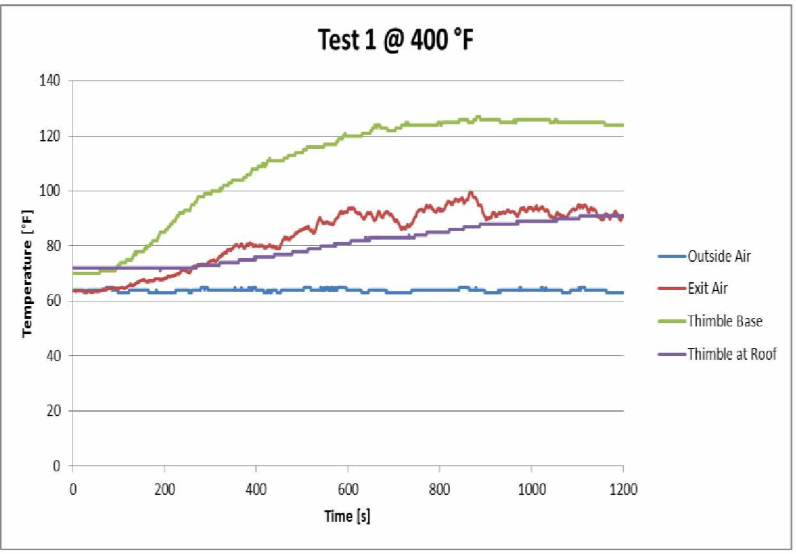

Figure 79: Cool Air Inlet Velocity was Approx. 105 ft/min

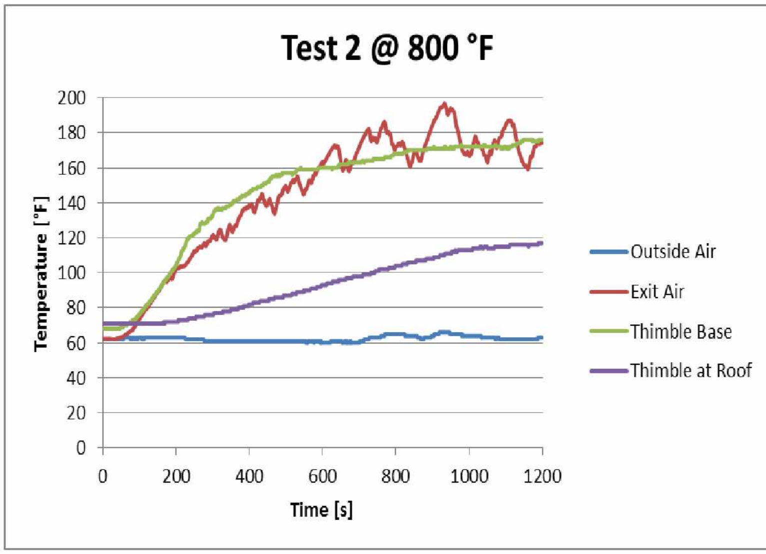

Figure 81: Cool Air Inlet Velocity was Approx. 130 ft/min

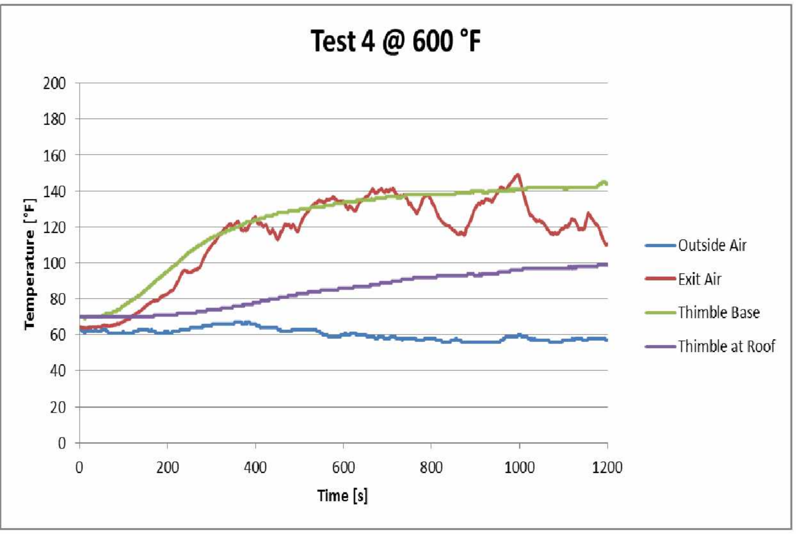

Figure 80: Cool Air Inlet Velocity was Approx. 115 ft/min

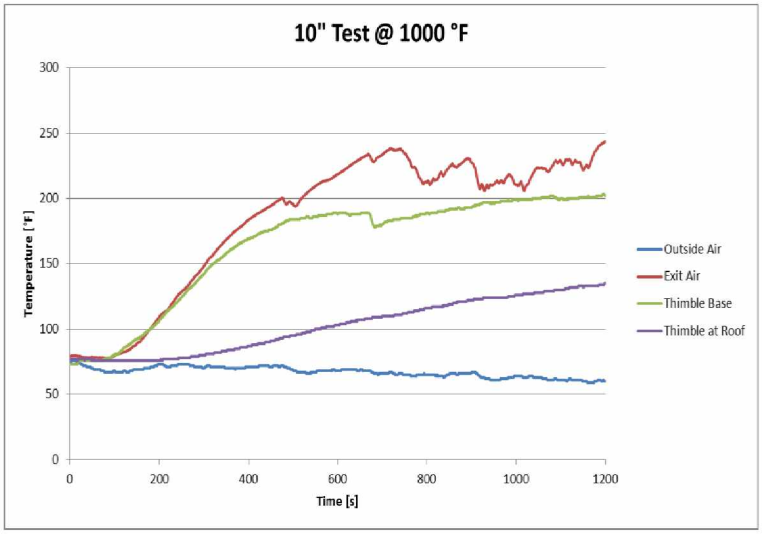

Figure 82: Cool Air Inlet Velocity was Approx. 145 ft/min

10in Thimble Summer Results

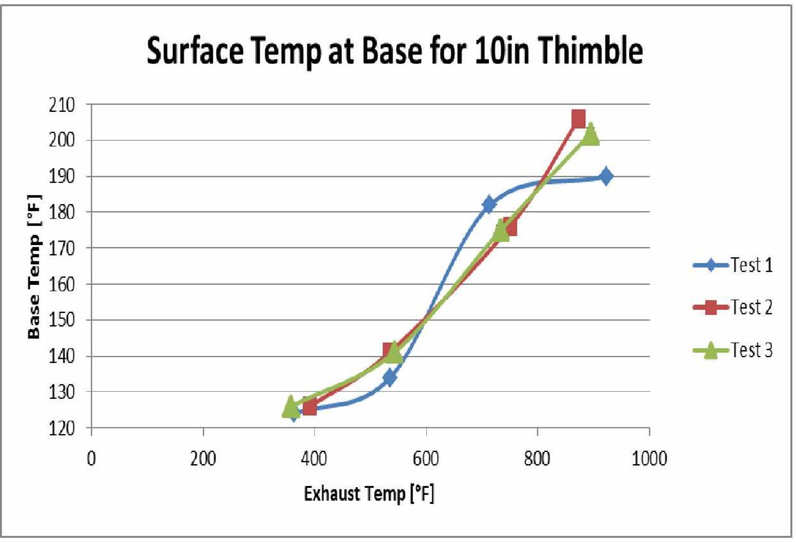

Figure 83: Surface Temperature at the Base of the Thimble

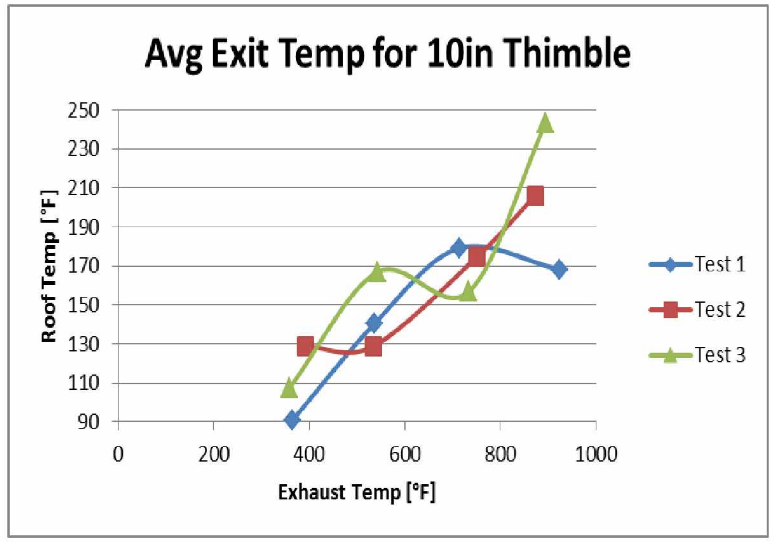

Figure 85: Exit Temperature of Cooling Air

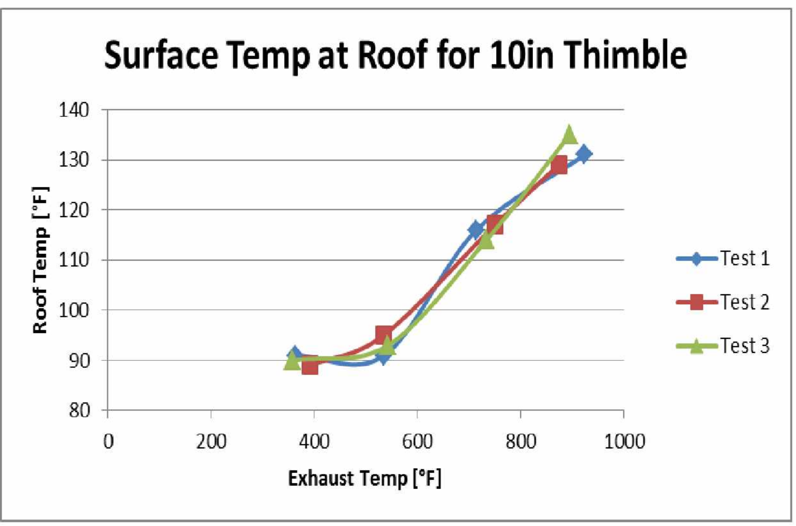

Figure 84: Surface Temperature Contacting Combustible Material

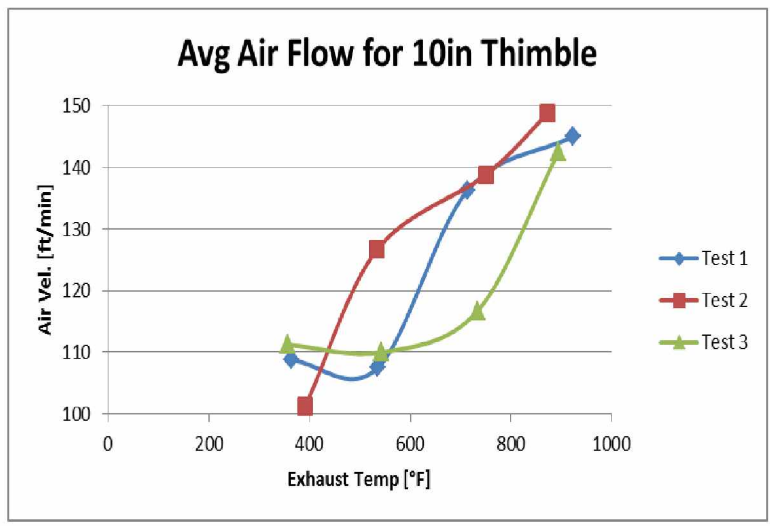

Figure 86: Cooling Air Inlet Velocity

10in Thimble Winter Results

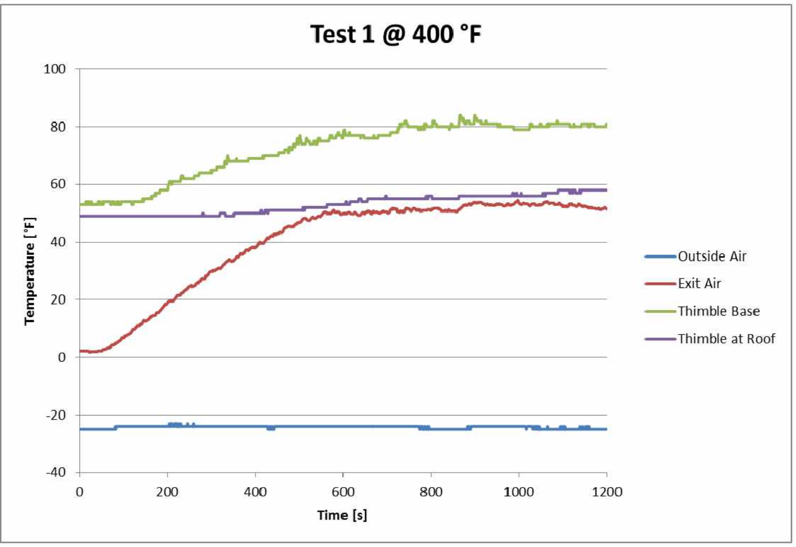

Figure 87: Cool Air Inlet Velocity was Approx. 300 ft/min

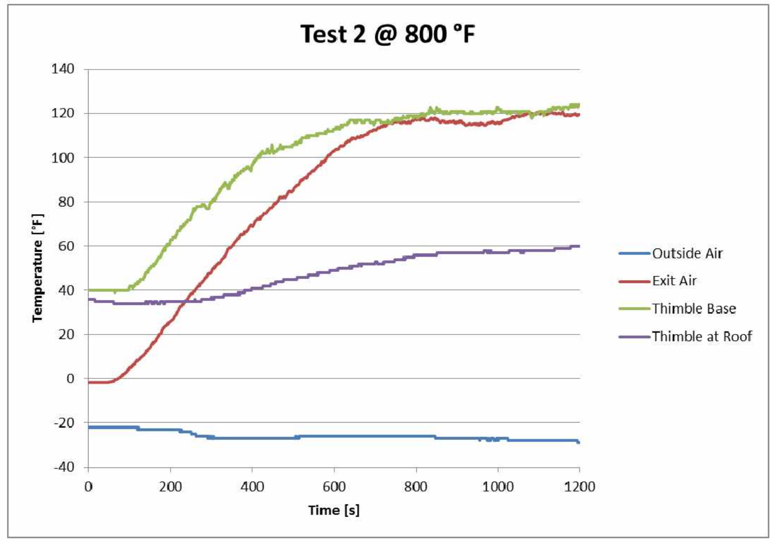

Figure 89: Cool Air Inlet Velocity was Approx. 220 ft/min

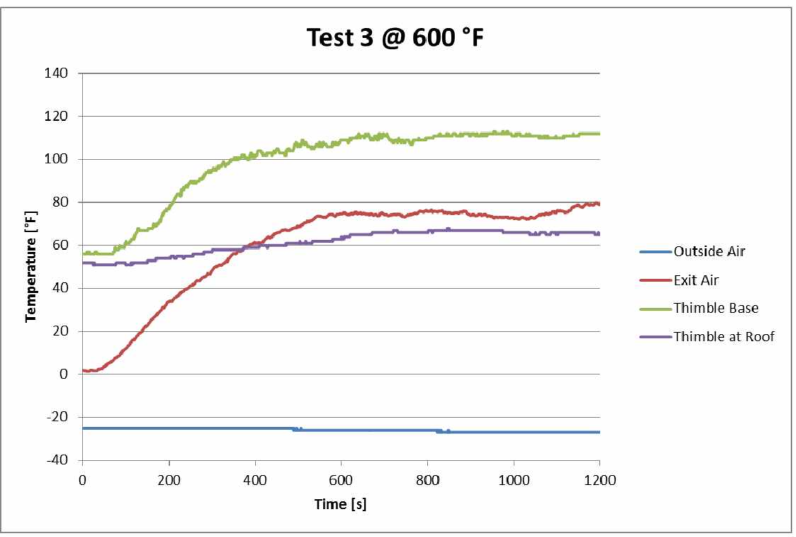

Figure 88: Cool Air Inlet Velocity was Approx. 260 ft/min

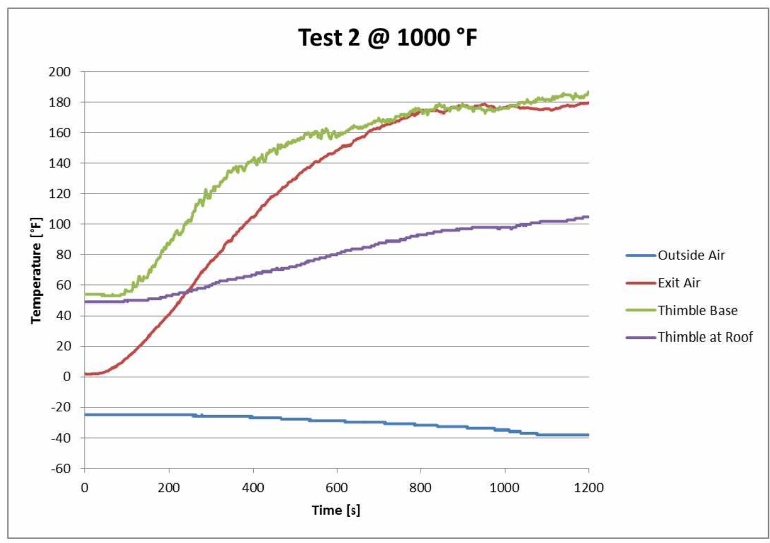

Figure 90: Cool Air Inlet Velocity was Approx. 160 ft/min

10in Thimble Winter Results

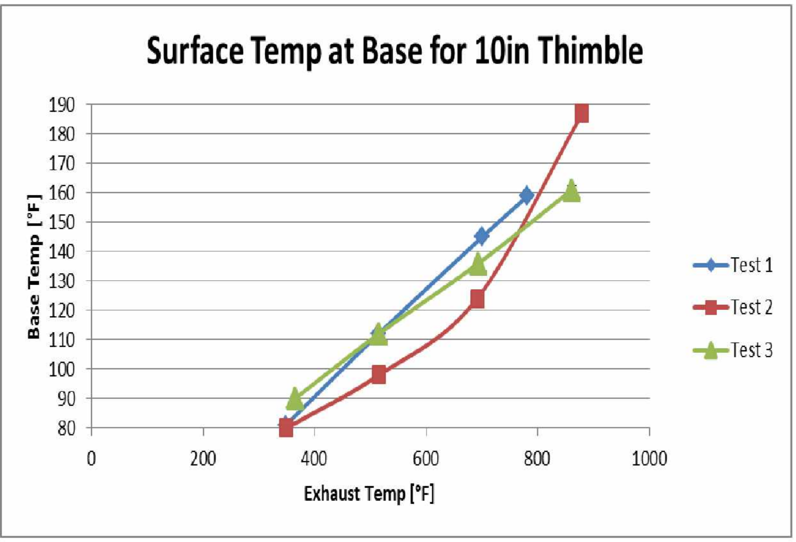

Figure 91: Surface Temperature at the Base of the Thimble

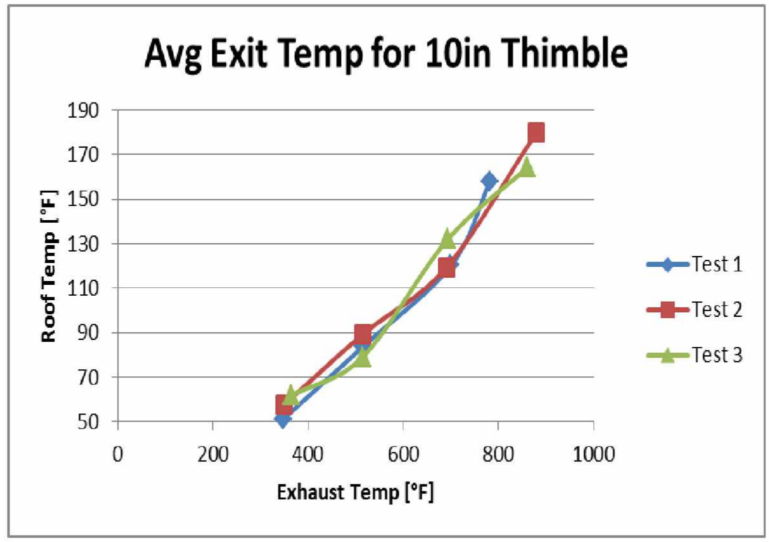

Figure 93: Exit Temperature of Cooling Air

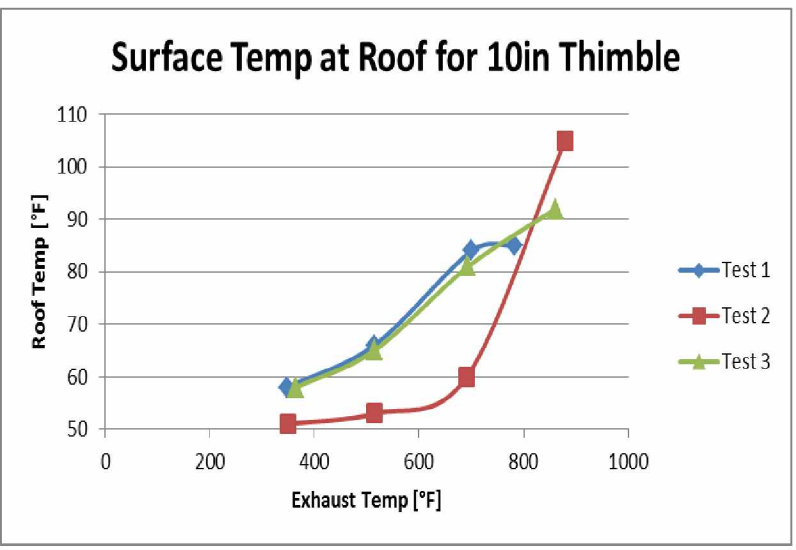

Figure 92: Surface Temperature Contacting Combustible Material

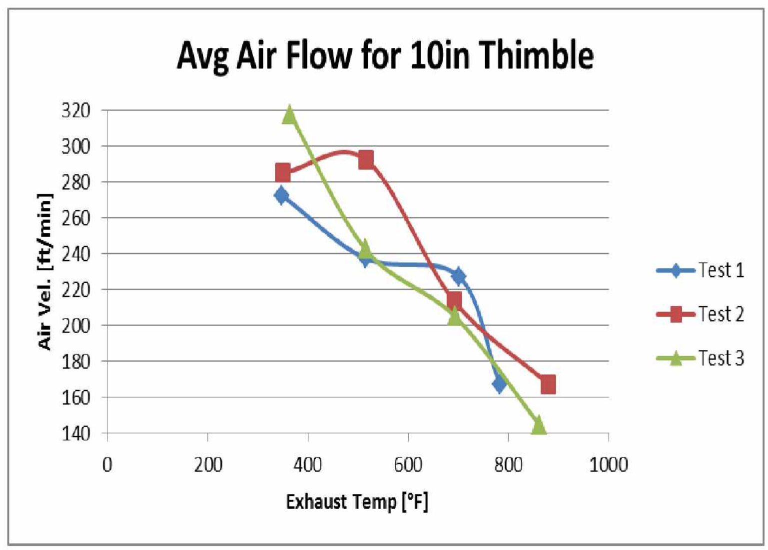

Figure 94: Cooling Air Inlet Velocity